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WAVE-EXCITED VIBRATION IN SHIPS—A MEANS OF CONTROLLING IT

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WAVE EXCITED VIBRATION IN SHIPS—A MEANS OF CONTROLLING IT

by K. V. TAYLOR, B.Sc., C.Eng.

SUMMARY

Wave-induced vibration has, in recent years, come more into prominence, and this paper discusses the nature of the phenomenon and the influence which it has on the service performance of ships. Also, the possible effect on their overall strength throughout their lives.

Various remedial solutions to reduce vibrations are discussed and, in particular, the use of a tuned tank as a vibration damper. This type of device has been proposed and tested by the National Physical Laboratory.

Some basic theory is given and calculations made to show the characteristic behaviour of such a device and the improvements expected with it in operation. Similar calculations are made to indicate its effectiveness to reduce the vibration following a slam.

1. INTRODUCTION

Wave-excited vibration is not a new phenomenon and has, no doubt, been experienced by the crews of ships since iron ships were constructed. Whether wooden ships can be included earlier than this is problematical but the physical nature of wood as compared with iron or steel would very significantly affect the hull response. Many references can be found in earlier naval architecture papers to flexural vibrations occurring as a result of a slam or wave impact in heavy seas, but a more specific comment is obtained in a paper by Dieudonne (1) in 1958.

This type of flexural vibration is similar to that experienced by a slender beam after a sudden blow, resulting in a multiplicity of mode shapes dependent on their natural frequency (Fig. 1). In practice, the damping of motion varies according to the number of nodes, and also the natural frequency, so that the higher modes are damped out first with the fundamental mode (2 node-mode) being sustained the longest. The amplitudes decay in a manner shown in Fig. 2 which conforms to a logarithmic decrement (2).

The introduction of all-welded ships has resulted in the hull structure having much less damping than that experienced by riveted construction, because of the friction loss which occurs with the straining of the plate joints. This has brought about a marked increase in flexural response to a slam, but also in a new range of flexural vibrations phenomena which have come to be known as 'Springing' and which are not dependent on slamming or severe weather conditions. It is difficult to ascertain the earliest reference to springing, and while attention was drawn to the phenomena in 1963 by the Author (3), an earlier reference was given relating to similar behaviour in craft on Russian inland waters (4).

It is obvious that a floating slender beam appropriate to that of a ship's main hull girder could be excited into vibrations with the passage of waves at a frequency coincidental with the natural frequency of one of the hull modes. In the case of an actual short-crested sea the frequency of encounter is not constant and the simple concept for excitation is invalid. However, if one assumes the wave energy spectrum to be made up of a large number of waves of different frequencies, combined to produce the real sea state, then it is possible for the energy in the particular frequency domain of the flexural vibration to excite the ship. This concept was used by Goodman (5) to provide a means of calculating the springing response of ships to unidirectional waves. Similar methods on the theme have been developed by van Gunsteren (6) and Kumai (7).

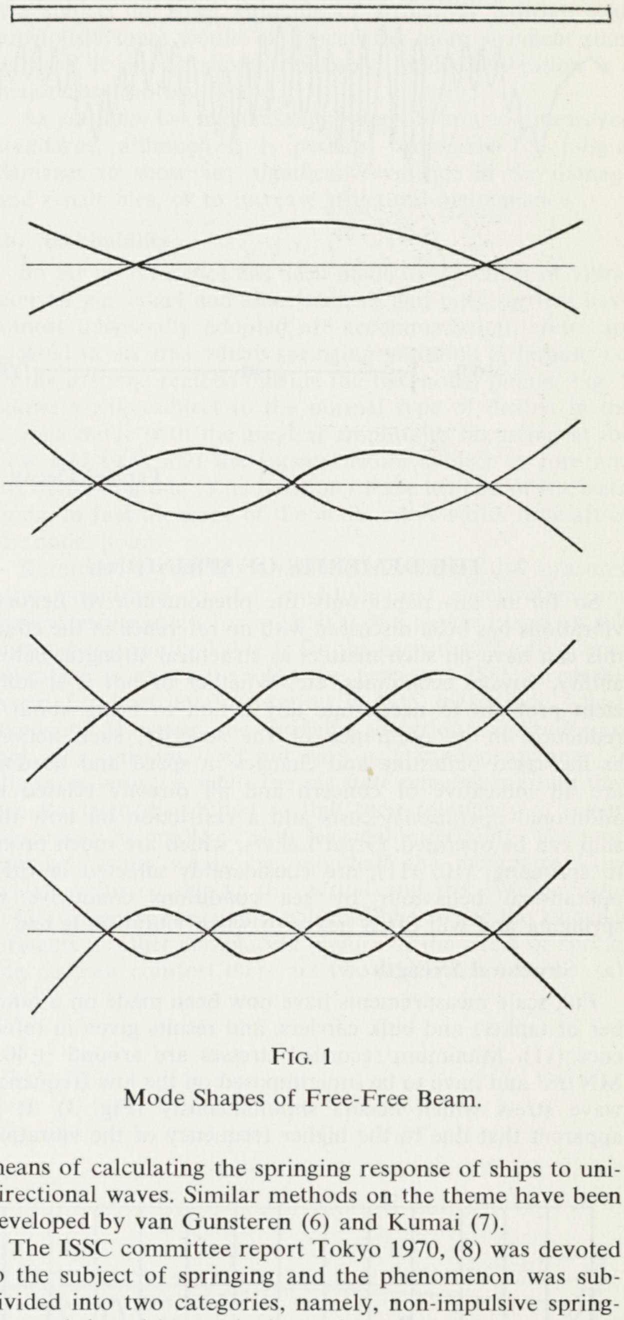

FIG. 1

Mode Shapes of Free-Free Beam.

The ISSC committee report Tokyo 1970, (8) was devoted to the subject of springing and the phenomenon was subdivided into two categories, namely, non-impulsive springing and impulsive springing. The former relates to the behaviour explained above and occurs solely due to the regular fluctuation of the component hydrodynamic pressure around the hull. The latter, however, occurs as a result of irregular impulses of a minor nature (not slams) often associated with light to moderate pitching. In certain conditions vibration can reach significant levels and be sustained over long periods of time. Reference 9 contains data which are appropriate to this latter type of springing and is generally associated with the small type of tanker and bulk carrier. Larger ships are less prone to pitching in moderate sea conditions and with lower fundamental 2-node mode frequencies are more likely to experience non-impulsive springing behaviour.

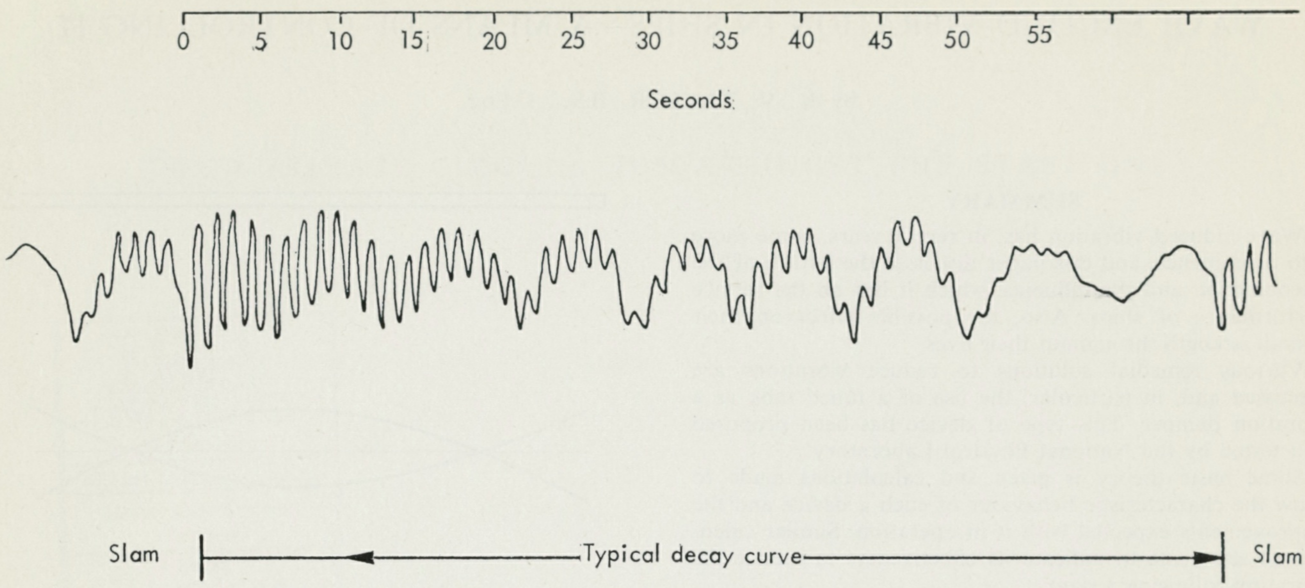

FIG. 2

Typical Decay of Vibration Response.

2. THE DEMERITS OF SPRINGING

So far in this paper only the phenomenon of flexural vibrations has been discussed with no reference to the effect this can have on such matters as structural strength, habitability, voyage economics, etc. Whether or not it is sufficient problem to necessitate any action to bring about a reduction in its occurence or the severity, such actions as increased ballasting and changes in speed and heading are all indicative of concern and all directly related to additional operational costs and a restriction on how the ship can be operated. Great Lakers, which are much prone to springing, (10) (11), are considerably affected in their operational behaviour by sea conditions conducive to springing and will often heave to when vibration is bad.

(a) Structural Strength

Full-scale measurements have now been made on a number of tankers and bulk carriers, and results given in reference (11). Maximum recorded stresses are around ± 40.8 MN/m^2 and have to be superimposed on the low-frequency wave stress which occurs simultaneously (Fig. 3). It is apparent that due to the higher frequency of the vibration response the number of cycles occurring in a given time is on an average of 3–4 times that caused by the passage of the waves themselves. Since the stress levels are relatively low it is in the region of fatigue that their influence would be most felt. A recent theoretical investigation on the effect of springing stress in tankers has been carried out by the Society (12). In this instance, calculated long-term wave and springing stress data were used in conjunction with S-N fatigue data for welded joints (13) to obtain cumulative damage employing the Miner's law concept. Standard cumulative distributions for wave and springing stresses were used to obtain comparative accumulative damage assessments and Fig. 4 illustrates how damage increases with increase in springing stress. The ordinate scale represents damage and a value of 1.86 is the damage equivalent to a mild steel ship with no springing (the particular detail being an F type weld).

Obviously this idealized concept relating only to the main hull girder connections simplifies the problem and does not enable a wider view to be made of all parts of the structure particularly whether there are significant local stress concentrations. Logically the structural designer faced with two

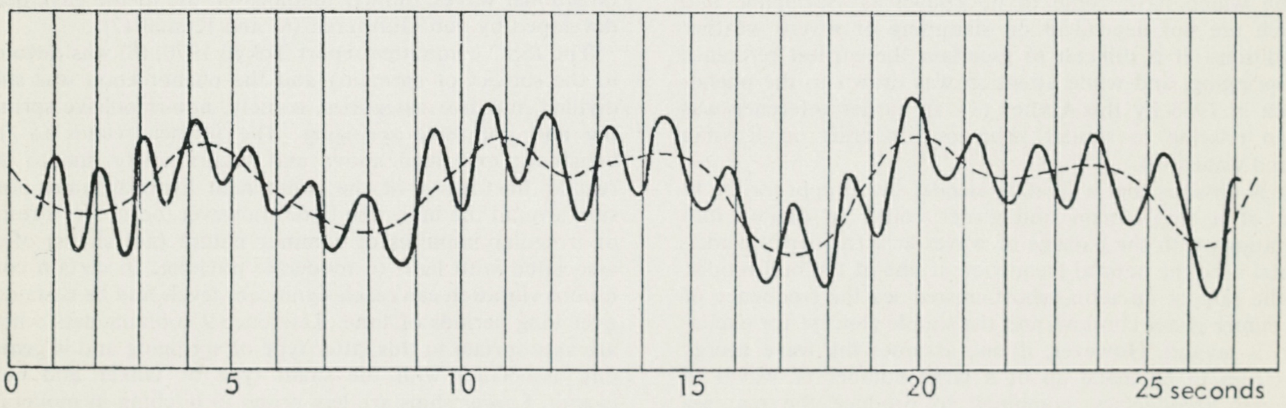

FIG. 3

The Combination of Low-Frequency Wave Stress and Springing Stress.

identical structures, one subject to the long periods of flectural vibrations over and above the normal loading while the other has no augmented stresses would instinctively question the concept of equal scantlings, that is, either increase the strength of the structure with vibration or reduce for the one where the vibration is absent.

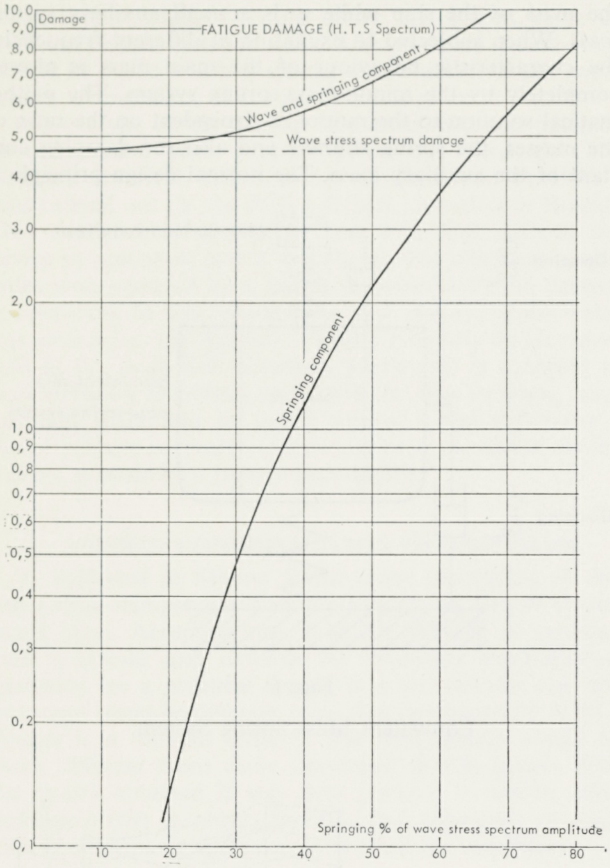

FIG. 4

Accumulative Fatigue Damage in Relation to Increase in Springing Stress.

The classification societies, upon whose responsibility the decision lies with respect to ships, are not unaware of this problem and have the matter very much in mind.

If existing standards of structural design are adequate in all ships subject to springing stress, there is perhaps a valid argument to accept lighter scantlings in less severe standards for a ship which is vibration free. Alternatively, if ships are subject to large amounts of structural damage than previously there would be a need for more stringent standards if it can be shown that wave-induced vibration is a major contributory factor.

As yet there has been no suggestions of any new improved standards, although it is perhaps too early for fatigue damages to show any significant evidence in the damage and repair files, or to increase structural maintenance.

(b) **Habitability**

So far no reference has been made to the effect of vibration on personnel and since tankers and bulk-carriers have almost universally adopted aft accommodation, crews are located in an area where springing vibration is largest, i.e. in the extreme regions outside the two nodal points. Fig. 5 shows a ship subject to the normal type of flexing in the 2-node mode with the greatest amplitudes occurring at the bow and stern and the superstructure subject to fore and aft oscillation due to its location on the aft end of the stern node. In fact the slope of the nodal curve varies little aft of the nodal point.

Reference 9 contains a method of relating the measured stress amidships to the amplitude and acceleration and typical information for a 250 000-dwt tanker based on this method as given in Table 1. On the basis that the frequency vibration is about 0·35 Hz the proposed ISO acceptable norm for horizontal vibration is 0·35 m/sec for 4-hour exposure. However, the springing vibration amplitudes are not at a constant level but vary in a random manner. In the short-term, i.e. while the sea state remains constant they are Rayleigh distributed so that their frequency of occurrence may be predicted with reasonable certainty. The most frequent value would be only half the maximum value which occurred during the period and the average value 0·625 times the maximum value. While so far the evidence presents a rather complacent picture of the effect of springing on crew comfort there are two additional factors.

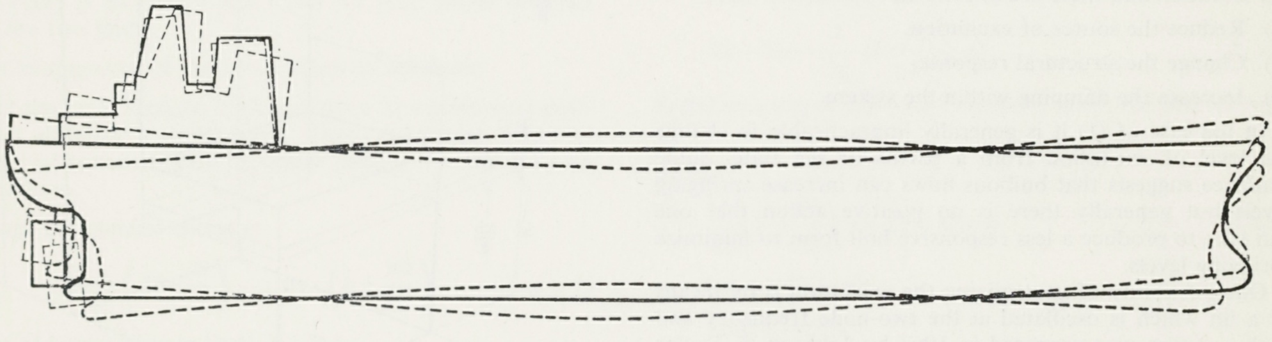

FIG. 5

Illustration Two Node Vertical Vibration of Ship.

TABLE 1

Typical values of stress, amplitude and acceleration for a 250 000-dwt tanker.

Springing Stress Amidships	$\pm$41 MN/m^2
Bow amplitude	$\pm$12·7 cms
Bow acceleration	$\pm$0·87 m/sec^2
Bridge amplitude	$\pm$6·1 cms
Bridge acceleration	$\pm$0·42 m/sec^2

Firstly, human susceptibility to vibration of this frequency has not received as much attention as those of higher frequencies, and whilst the ISO proposals (14) suggest that the body does not react adversely to vibration of this nature, it is felt that more research is required on this subject. Secondly, springing vibration is in addition to propeller and often machinery vibration which are by no means insignificant in themselves and in many cases are at a level which are unsatisfactory. Human response to several simultaneous sources of vibration is even more unpredictable than to a single source, and the Author as one who has experienced the sensation on a number of occasions can adequately appreciate the discomfort.

(c) **Operational Problems**

Lastly, on the demerits of springing there are the effects of those actions to reduce springing on the normal operation of the ship. As indicated earlier these pertain either to increased loading, speed reduction and heading alterations. Springing is normally a ballast problem and rarely, when the ship is fully loaded. To reduce the level of springing in large ships many companies are using ballast conditions with much deeper draughts than would be required for the sea conditions. For example, in a 250 000-ton tanker the speed loss with 55 per cent ballast instead of 35 per cent ballast would be about 5 per cent.

For changes of speed and heading the effects are much less certain and generally more unpredictable, in the effect on the springing it could be argued that any engine or course changes made over and above those required in normal navigation must constitute a loss of efficiency.

3. THE SIMPLE TUNED-MASS SYSTEM

There are several methods generally available for reducing vibration and these are as follows:—

(1) Reduce the source of excitation.

(2) Change the structural response.

(3) Increase the damping within the system.

In the case of (1) it is generally impracticable to change the level of excitation from a particular sea state. Some evidence suggests that bulbous bows can increase springing levels but generally there is no positive action that one can take to produce a less responsive hull-form to minimize springing levels.

One possibility of neutralizing the excitation is by means of a fin which is oscillated at the two-node frequency and such a device was proposed in 1963 by Johnson & Taylor (15). Fin angle, however, would be required to be changed to take account of pitching with forward speed, this necessitating an elaborate control mechanism.

As regards (2), no change large enough to increase the fundamental frequency into a region where there is much less wave energy for excitation is practicable. Finally, the third possibility, namely to increase the damping of the main hull, might have some merit particularly as riveted construction did precisely that.

A further alternative combining (2) and (3), however, does exist, namely the tuned damper. A device on this principle was used as far back as 1924 (16) and a paper on the theory of the vibration absorber was read in 1928 (17).

The principle is simply illustrated in Fig. 6. M represents the mass of the ship while m is a small auxiliary sprung mass. When subjected to excitation at different frequencies the characteristic behaviour of the main mass is altered completely by the minor mass-sprung system. The mathematical solution to the motion is dependent on the ratio of the masses, the spring stiffness and also the damping constant of the auxiliary mass. The normal design principle is

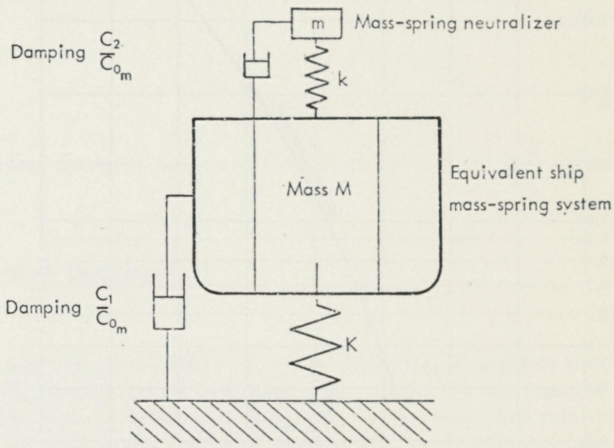

Fig. 6

Equivalent Mass Spring System.

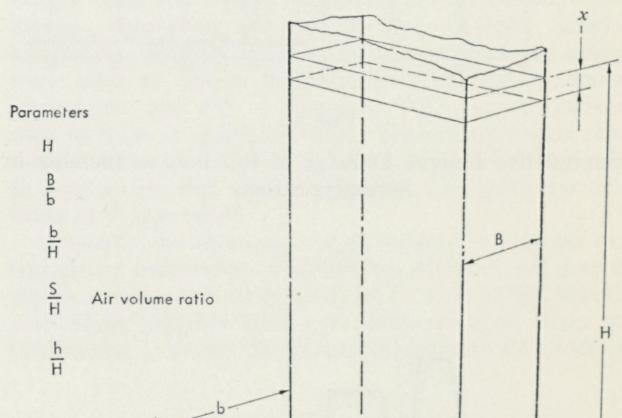

Fig. 7

Diagrammatic Arrangement of Damper.

to ensure that the two resonant conditions are of equal magnitude.

It is, perhaps, not surprising that this technique could be utilized to provide a simple solution to the springing problem—except that any mechanical device of the necessary proportions would be extremely cumbersome and expensive. It is at this point that the National Physical Laboratory Ship Hydrodynamic Division under Mr. Paffett came up with an elegant solution (18). The auxiliary tuned mass should be liquid which is sprung on a cushion of air, and a typical arrangement of such a device is given in Fig. 7.

A prototype of the damper was developed and patented in conjunction with support given by N.R.D.C. (The National Research Development Council) and in 1970 a trial carried out on the frigate H.M.S. *Scorpion* in Rosyth. This consisted of an exciter test with and without the device in operation and it was found that vibration amplitudes were reduced by a factor of seven with the damper in operation. In this instance, the tank containing the water was external to the hull but it would normally be incorporated in the main hull structure, preferably in a region of high vibration amplitude such as at the bow or stern. Also, the tank space and its liquid content could still serve a useful purpose as reserve fresh-water or, if the liquid chosen was oil, it could be a reserve fuel supply.

4. DESIGN CONSIDERATION OF THE TANK

As indicated in Section 3, the main innovation of this tuned-mass damper has been the concept of the air-sprung liquid mass. Accepting this, it is not difficult to envisage such a system and develop the necessary mathematical treatment for a practical design. It is pointed out that this treatment could be different from that developed by N.P.L., though it is thought unlikely that their results would be much different from those developed in this paper. With the results obtained it was then possible to utilize other techniques (19) to obtain the modified behaviour of ships with this device in operation in various sea conditions.

(a) Design Considerations for the Tank

The basic device is shown in Fig. 7 and the following assumptions will be made: —

(1) The masses in the main and minor limbs have velocities $\dot{x}$ and $\frac{\dot{x}B}{b}$ respectively—actual hydrodynamic behaviour is not considered.

(2) There is no energy loss when the fluid passes between the two limbs.

(3) Compression of the air is taken as adiabatic.

Let the main limb be displaced from its equilibrium position a distance x downwards. Then, change in potential energy of system = gain in kinetic energy + work done on air cushion.

Change in potential energy=

$$\rho\, Bxc\left(H-\frac{x}{2}\right)-\rho\, Bxc\left(h+\frac{xb}{2B}\right) \qquad (4\ a.1)$$

Mass of liquid in main limb $=\rho\,$Bch

Mass of liquid in minor limb$=\rho\,$bch

x is small in relation to H and h.

$$\text{Gain in kinetic energy}=\frac{1}{2g}\rho\, BcH\dot{x}^2+\frac{1}{2g}\rho\, bch\left(\frac{\dot{x}B}{b}\right)^2 \qquad (4\ a.2)$$

$$\text{Work done on air cushion}=\frac{p_2v_2-p_1v_1}{\gamma-1} \qquad (4\ a.3)$$

Initial pressure $p_1=\rho\,(H+h_a-h)-\rho\, h_a=$atmospheric pressure

$$\text{pressure } p_2=p_1\left(\frac{v_1}{v_2}\right)^{\gamma}$$

Let v_1=sbc and v_2=sbc–xBc

$$p_2=\rho\,(H+h_a-h)\left(\frac{sbc}{sbc-xBc}\right)^{\gamma} \qquad (4\ a.4)$$

Substituting in (4 a.3)

Work done=

$$\frac{\rho\,(H+h_a-h)\left(\frac{sbc}{sbc-xBc}\right)^{\gamma}(sbc-xBc)-\rho\,(H+h_a-h)\,sbc}{\gamma-1}$$

$$=\rho\,\frac{(H+h_a-h)}{\gamma-1}sbc\left[\left(1-\frac{xB}{sb}\right)^{-\gamma+1}-1\right] \qquad (4\ a.5)$$

$$\text{Expanding}\left(1-\frac{x}{s}.\frac{B}{b}\right)^{1-\gamma}=$$

$$\left[1-(1-\gamma)\frac{x}{s}\frac{B}{b}+\frac{(1-\gamma-1)(1-\gamma)}{2}\frac{x^2}{s^2}.\frac{B^2}{b^2}-1\right]$$

$$=\left[(\gamma-1)\frac{B}{b}\cdot\frac{x}{s}+\frac{(\gamma-1)\gamma}{2}\frac{B^2}{b^2}\cdot\frac{x^2}{s^2}\right]$$

Work done on air=

$$\frac{\rho\,(H+h_a-h)}{(\gamma-1)}sbc\left[(\gamma-1)\frac{B}{b}\frac{x}{s}+\frac{(\gamma-1)}{2}\gamma\frac{B^2}{b^2}\frac{x^2}{s^2}\right] \qquad (4\ a.6)$$

neglecting higher powers of x

Equation of conservation of energy

Loss of potential energy = gain of kinetic energy + work done on air

$$\rho\, Bxc\left(H-\frac{x}{2}\right)-\rho\, Bxc\left(h\frac{xB}{2b}\right)=\frac{1}{2g}\rho\, BcH\dot{x}^2+\cdot$$

$$\frac{1}{2g}bch\left(\frac{B\dot{x}}{b}\right)^2+\rho\,\frac{(H+h_a-h)}{\gamma-1}sbc\left[(\gamma-1)\frac{Bx}{bs}+\left(\frac{\gamma-1}{2}\right)\frac{\gamma B^2}{b^2}\frac{x^2}{s^2}\right] \qquad (4\ a.7)$$

Divide equation (4 a.7) by $\rho\,$cBh

$$\text{and putting }\frac{h_a}{H}=X\text{ and }\left[\frac{1}{2H}+\frac{B}{2Hb}+\frac{(H+h_a-h)}{2H}\frac{\gamma B}{b}\cdot\frac{1}{S}\right]=Y$$

Thus equation (4 a.7) becomes

$$Xx-Yx^2=\frac{1}{2g}\left(1+\frac{Bh}{bH}\right)\dot{x}^2 \qquad (4\ a.8)$$

Differentiating (4 a.8) with respect to t and dividing through by $\dot{x}$.

giving

$$\frac{1}{g}\left(1+\frac{Bh}{bH}\right)\ddot{x}+2Yx-X=0 \qquad (4\ a.9)$$

Let u=2Yx−X

Then $\ddot{u}=2Y\ddot{x}$

$$\frac{\frac{1}{g}\left(1+\frac{Bh}{bH}\right)}{2Y}\ddot{u}+u=0 \qquad (4\ a.10)$$

$$\omega^2=\frac{2Y}{\frac{1}{g}\left(1+\frac{Bh}{bH}\right)}$$

The tank frequency

$$f=\frac{\omega}{2\pi}=\frac{60}{2\pi}\sqrt{\frac{g}{H}\cdot\frac{\left[1+\frac{B}{b}+(H+h_a-h)\cdot\frac{\gamma B}{bs}\right]}{\left(1+\frac{Bh}{bH}\right)}}\ \text{(cpm)} \tag{4 a.11}$$

Equation (4 a.10) has been examined with respect to the parameter $\frac{B}{b}$, $\frac{h}{H}$, $\frac{s}{H}$ and typical plots are presented in Figs. 8 and 9. Fig. 8 shows the variation in the natural frequency of the tank for varying height of water column with different $\frac{h}{H}$ and $\frac{s}{H}$ parameters, $\frac{B}{b}$ being a different ratio fixed for each figure. Height of water column has the most marked effect on frequency.

Variation of frequency with the ratio of water column to reservoir breadth is shown in Fig. 9 and this supplements the information already contained in the previous diagram.

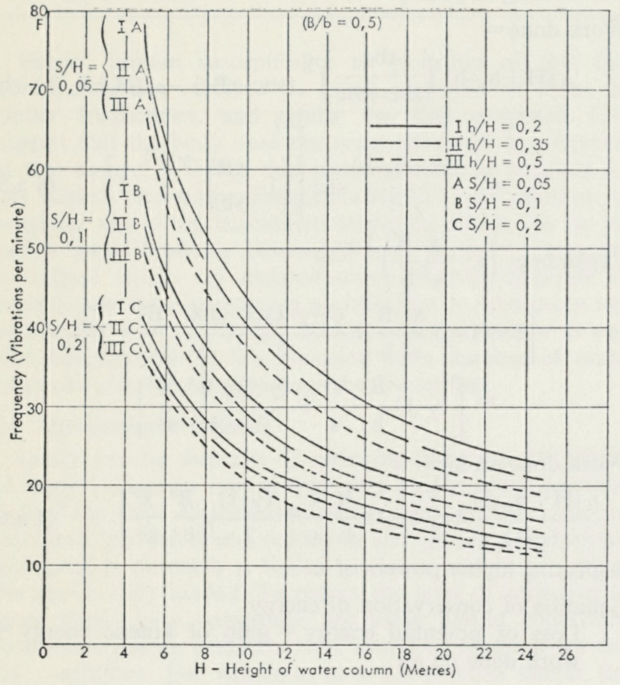

FIG. 8

Frequency of Vibrations with Varying Heights of Water Column for Different Parameters of Vibration Damper.

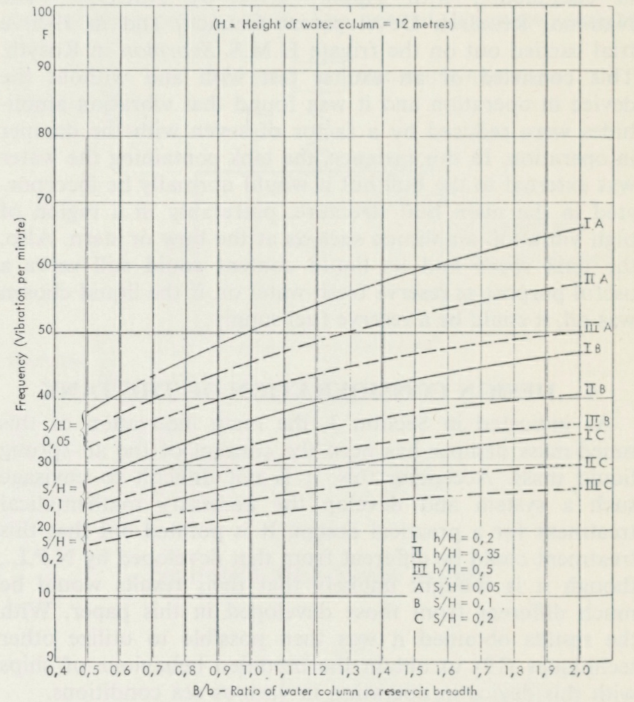

FIG. 9

Frequency of Vibrations with Varying Water Column to Reservoir Breadth Ratio.

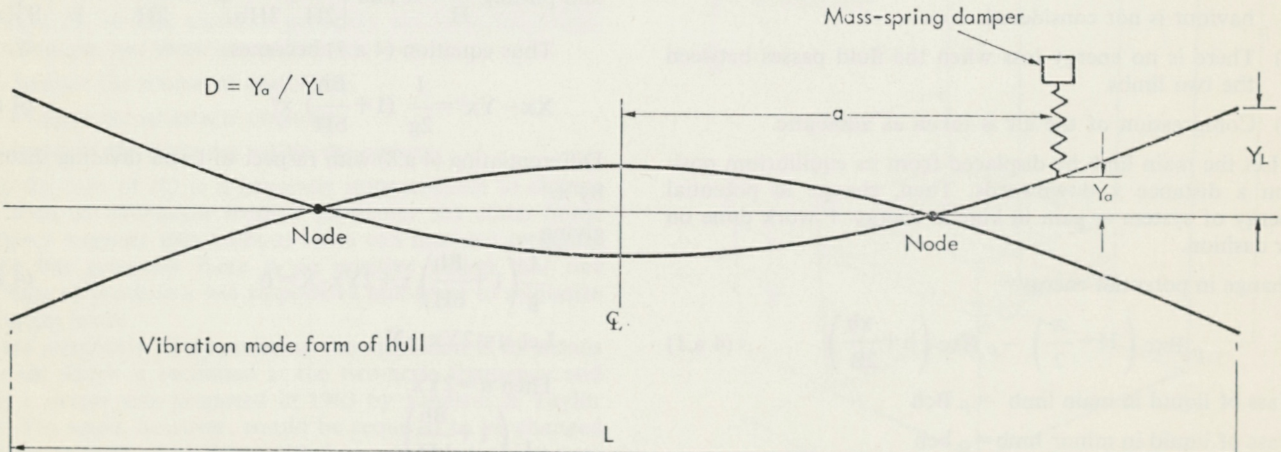

FIG. 10

Diagrammatic Arrangement of Damper in Ship.

(b) **Vibration Response of Ship and Damper System**

Fig. 10 shows an idealized arrangement of the tuned damper located on a vibrating hull profile, where L is the length of the ship and a is the distance of the exciter from amidships. Y_L is the half amplitude of vibration at the ends of the ship and Y_a is the corresponding half-amplitude of vibration of the hull at the damper location. $D = Y_a/Y_L$ varies from 1 at the ends of the ship to 0 at the nodes and approximately 0·30 (20) midway between.

The equivalent vibration system is shown in Fig. 6 where the ship is represented by mass M supported on a spring of stiffness K. The damper device is likewise represented by mass m, supported on a spring of stiffness k, but located on mass M.

The natural frequency of the M–K system is the 2-node mode frequency of vertical vibration of the ship. The natural frequency of the smaller m–k system varies with the parameters used in Section 4(a) and may be adjusted to have any desired natural frequency value close to that of the large M–K system.

The damping of the main system which represents the main hull is represented by the damping dash-pot C_1 and the damping of the device by the dash-pot C_2. This latter dash-pot is located between m and M and therefore reflects the relative velocity motion between the ship and water in the damper.

Consider M and m to be displaced x_1 and x_2 respectively and let force $P_o \sin \omega t$ act on mass M.

The equations of motion for the masses M and m are respectively: —

$$M\ddot{x}_1 + Kx_1 + k(x_1 - x_2) + C_1\dot{x}_1 + C_2(\dot{x}_1 - \dot{x}_2) = P_o \sin \omega t \quad (4\ b.1)$$

$$m\ddot{x}_2 + k(x_2 - x_1) + C_2(\dot{x}_2 - \dot{x}_1) = 0 \quad (4\ b.2)$$

Equations (2 b.1) and (2 b.2) can be put into the following forms.

$$-M\omega^2 \bar{x}_1 + k\bar{x}_1 + k(\bar{x}_1 - \bar{x}_2) + j\omega C_2(\bar{x}_1 - \bar{x}_2) + j\omega C_1\bar{x}_1 = P_o \quad (4\ b.3)$$

$$-m\omega^2\bar{x}_2 + k(\bar{x}_2 - \bar{x}_1) + j\omega C_2(\bar{x}_2 - \bar{x}_1) = 0 \quad (4\ b.4)$$

where $\bar{x}_1$ and $\bar{x}_2$ are (unknown) complex numbers and $x_1 = I(\bar{x}_1 e^{j\omega t})$ $x_2 = I(\bar{x}_2 e^{j\omega t})$

Bringing together the terms in $\bar{x}_1$ and $\bar{x}_2$

$$[-M\omega^2 + K + k + j\omega(C_1 + C_2)]\bar{x}_1 - [k + j\omega C_2]\bar{x}_2 = P_o \quad (4\ b.5)$$

$$-[k + j\omega C_2]\bar{x}_1 + [-m\omega^2 + k + j\omega C_2]\bar{x}_2 = 0 \quad (4\ b.6)$$

Equations (4 b.5) and (4 b.6) can be solved for $\bar{x}_1$ and $\bar{x}_2$

$$\bar{x}_1 = P_o \frac{(k - m\omega^2) + j\omega C_2}{[(M\omega^2 - K)(m\omega^2 - k) - m\omega^2 k - \omega^2 C_1 C_2] + j\omega C_2\left[-M\omega^2 + K + \frac{C_1}{C_2}k - \frac{C_1}{C_2}m\omega^2 - m\omega^2\right]}$$

This is in the form

$\bar{x}_1 = P_o \frac{(A + jB)}{(C + jL)}$ which can be transformed as follows

$\frac{(\bar{x}_1)}{P_o} = \sqrt{\frac{A^2 + B^2}{C^2 + D^2}}$ where $(\bar{x}_1)$ is the length of the $\bar{x}_1$ vector (4 b.7)

Applying this to x_1 we have $x_1 = I(\bar{x}_1 e^{j\omega t})$

or $x_1 = \bar{x}_1 \sin(\omega t + \alpha)$ where $\tan \alpha = \frac{BC - AD}{AC + BD}$

$\bar{x}_1$ being the vibration amplitude of the hull

$$\frac{(\bar{x}_1)}{P_o} = \sqrt{\frac{(k - m\omega^2)^2 + \omega^2 C_2^2}{\left[(m\omega^2 - k)(m\omega^2 - K) - m\omega^2 k - \omega^2 C_1 C_2\right]^2 + \left[-M\omega^2 + K + \left(\frac{C_1}{C_2}\right)k - \left(\frac{C_1}{C_2}\right)m\omega^2 - m\omega^2\right]^2 \omega^2 C_2^2}} \quad (4\ b.8)$$

Finally.

This can be reduced to non-dimensional form by dividing numerator and denominator by K^2

Also $x_{STATIC} = \frac{P_o}{K} = \frac{P_o}{M\Omega^2_M}$

$$\frac{(\bar{x}_1)}{P_o} = \frac{(\bar{x}_1)}{x_{ST} M \Omega_2^{\,2}} \quad (4\ b.9)$$

$$\frac{x}{x_{ST}} = \sqrt{\frac{\left[(f^2 - g^2)^2 + \left(2\frac{C_2}{C_{O_m}}\right) \cdot g^2\right]^2}{\left[(1 - g^2)(f^2 - g^2) - \mu g^2 f^2 - 4\left(\frac{C_1}{C_{O_M}}\right)\left(\frac{C_2}{C_{O_m}}\right)g^2 f\right]^2 + g^2\left(\frac{C_2}{C_{O_m}}\right)^2\left[-g^2 + 1 + \left(\frac{C_1}{C_2}\right)\mu f^2 - \left(\frac{C_1}{C_2}\right)\mu g^2 - \mu g^2\right]^2}}$$

where $\frac{x}{x_{ST}}$ = Dynamic magnifier

$\omega_a^{\,2} = \frac{k}{m}$ = Natural frequency of m–k system

$\Omega_M^{\,2} = \frac{K}{M}$ = Natural frequency of M–K system

$f = \frac{\omega_a}{\Omega_M}$; $g = \frac{\omega}{\Omega_M}$

$C_{O_M} = 2M\Omega_M$; $C_{O_m} = 2m\omega_a$

The typical response characteristics of the 2 mass-2 spring system are shown in Fig. 11. These are presented in the form of dynamic magnifiers x/x_o against non-dimensional frequency of excitation ω/Ω_M i.e. g

It will be seen that the single response for the fundamental 2-node mode frequency is transformed into a two-humped curve with a peak above and below the original

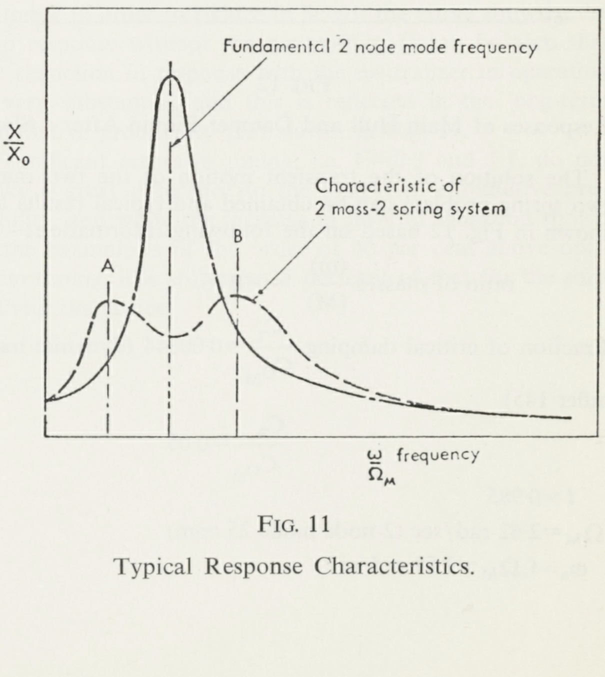

FIG. 11

Typical Response Characteristics.

single mass response frequency. The aim in selecting the appropriate parameters is to reduce the size of these two humps and to make them approximately equal magnitude. The peak below $\omega/\Omega_M = 1$ will be called A and that above called B. It will be shown later that altering the natural frequency of the damper ω_a, and thus f, will significantly change the location and magnitude of A & B.

5. RESPONSE TO TRANSIENT FORCES

Until now, the damper device has been considered only in relation to steady vibration, but what would the effect be if the ship was subject to a sudden shock as a result of a slam? Would the device be effective in altering the structural behaviour during the initial response period while at the same time quickly reducing the vibration which follows the slam.

As regards the severity, Meek (21) and Aertssen (22) have shown that the 2-node mode vibration can be at least twice as severe as that which occurs from wave-excited vibration, although it is rarer in occurrence and of shorter duration. Nevertheless, the damage caused by these reversals could be important both in relation to fatigue and from the avoidance of large accelerations of the main hull girder. It could equally be desirable to use the damper against the adverse effects of slamming as for minimizing the vibration from wave-excited vibration.

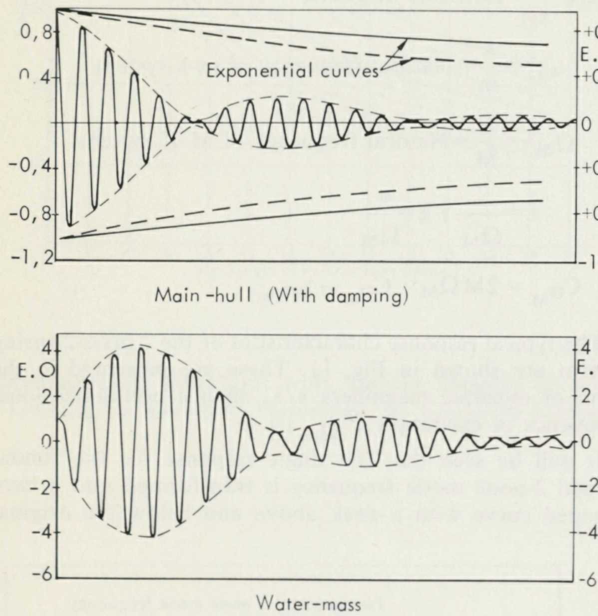

FIG. 12

Responses of Main Hull and Damper Liquid After a Slam.

The solution of the transient motion of the two mass-two spring problem can be obtained and typical results are shown in Fig. 12 based on the following information:—

$$\text{ratio of masses } \frac{(m)}{(M)} = 0{\cdot}0176.$$

Fraction of critical damping $\dfrac{C_1}{C_{O_M}} = 0{\cdot}00344$ (dynamic magnifier 145)

$$\frac{C_2}{C_{O_m}} = 0{\cdot}05$$

$f = 0{\cdot}985$

$\Omega_M = 2{\cdot}62$ rad/sec (2-node mode 25 cpm)

$\omega_a = f.\Omega_M = 2{\cdot}58$ rad/sec

It must be pointed out that the damping data in (2) indicate significantly larger log-decrement values than normally associated with the 2-node mode (steady state vibration). This is probably due to the presence of higher order modes which have much larger damping. For this exercise, however, the reduced value of damping was used, although both exponential curves are shown for comparison in the diagram.

It will be seen that compared with the undamped response as indicated by the exponential curves the 'damped' response is very considerably reduced as energy from the main mass (hull) is taken up by the minor mass (water). Some ten seconds later part of this energy is transferred back into the hull giving a short period of vibration with a much reduced amplitude. In a practical device it is quite likely that the initial motion of the water in the tank would create non-linear damping, which would dissipate the energy more rapidly and still further reduce the second-stage vibration. It is certainly a possibility that the damper design could incorporate such a feature.

6. PRACTICAL DESIGN APPLICATION

For a damper installation there are two main design requirements to be satisfied, namely, its working frequency and its water mass. As regards the operational frequency range to be covered by the device, this may be estimated for the ship for the desired service conditions and the frequency equation (4 a.11) used to determine the basic dimensions.

The second requirement to be considered is that concerning the size of tank necessary to provide an adequate effective mass capable of attenuating the induced vibration. This value will vary with the location of the tank with its design. An approximate figure for a suitable design would be between ½ per cent and 1 per cent of the ship's displacement. It should be pointed out, however, that this figure should not be confused with that based on the 'effective' masses involved, i.e. those necessary for an equivalent dynamic system.

It is now necessary to proceed along the lines described in Section 4(b) in order to examine the capability of the damper. As a preliminary check on its characteristics, the behaviour of the system is examined subject to varying excitation and constant force. For a typical example, consider a dynamic system where the ratio of the normalized masses (with respect to 2-node vertical) of the water in the tank and ship plus entrained water is 0·0176, with a corresponding damping $\dfrac{C_1}{C_0}$ of 0·00344. Consider the tank damping parameter $\dfrac{C_2}{C_0}$ to have values 0·025, 0·05 and 0·075 and the ratio (F) of the natural frequency of the tank to the ship to have values 0·96, 1·0 and 1·04.

The results of these calculations are shown in Fig. 13 and this indicates that the optimum condition for the tank occurs with a damping of 0·075 and an F value of 0·985. This shows a three-dimensional plot of 'peak' excitation against $\dfrac{C_2}{C_0}$ and F, the two ordinates at each position representing the maximum values of the two resonances, the full line joining the lower frequency value (peak A) and the dotted line the higher (peak B). These give two surfaces, the inter-section of which indicates a condition where the peaks are of equal magnitude. This line of intersection is at a roughly constant value of F=0·985.

A typical design of damping tank is shown in Fig. 14, and this would be suitable for a tanker of 200 000 dwt. capacity.

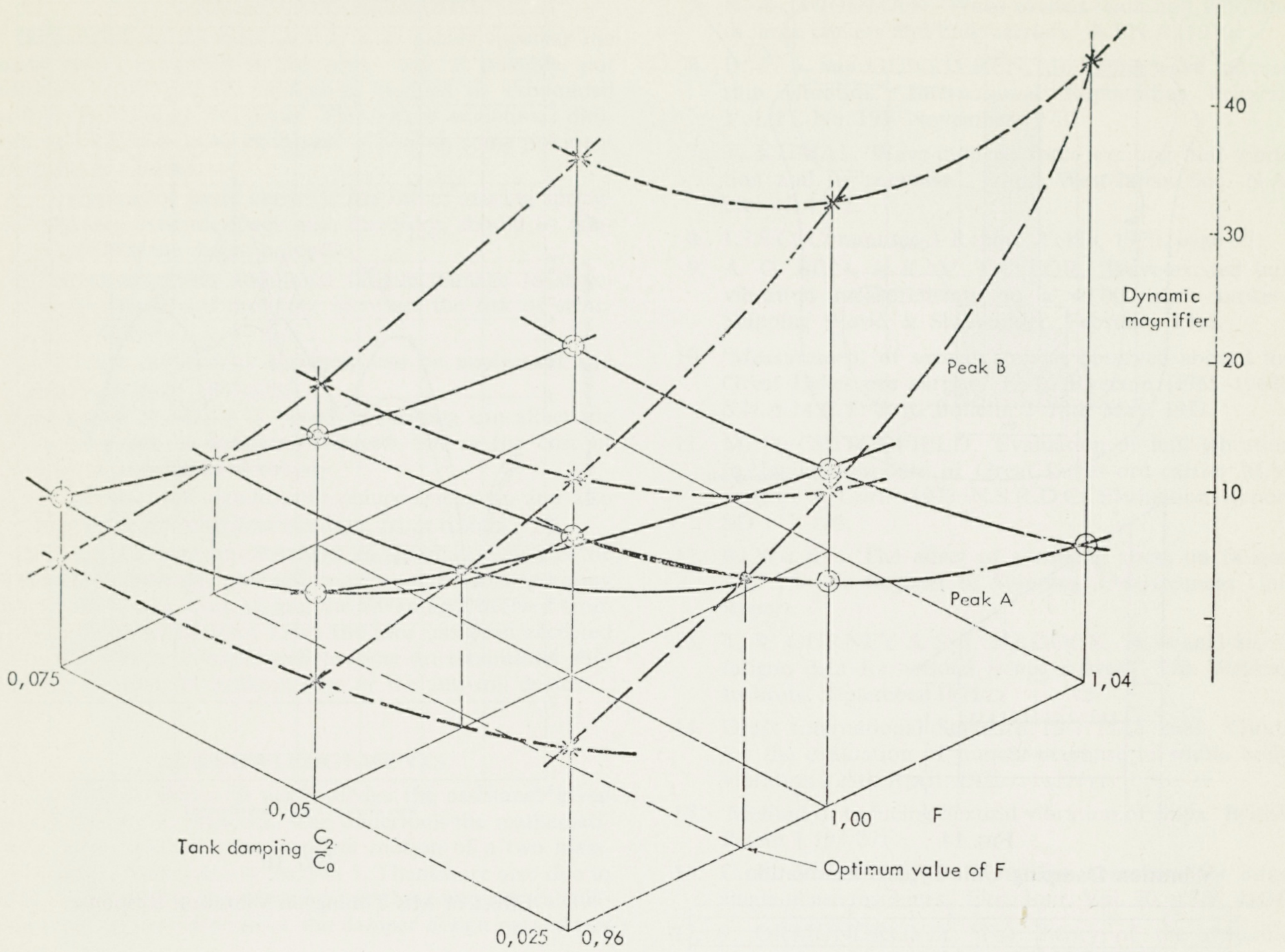

FIG. 13

Ship Response Operators for Varying Tank Damping.

In a paper to the Royal Institution of Naval Architects in 1970, Goodman (5) presented a theory of wave-excited vibration based on the dynamic response of a ship to cyclic buoyancy forces produced by waves. In this, the behaviour and response of the hull to a regular train of waves were extended to irregular seas and a long-term prediction of vibration stress determined. This work has been further developed and included in the Society's suite of programs (LR 241D and LR 310). The effect of the vibration damper on the vibration of the ship is obtained by modifying the vibration transfer function from which the non-dimensional variance of response is obtained. The basic method employed, once the transfer functions have been derived for varying encounter frequencies, is based on the Pierson-Moskowitz sea spectra.

Figs. 15 and 16 show typical curves of calculated non-dimensional bending moment response on a base of encounter frequency. The former compares the response with and without the vibration neutralizer in operation, while the latter figure shows the effect on the vibration response if the device is not 'tuned' correctly.

The ordinate scales of these plots have been made logarithmic in order to reduce in height the curve showing the ship response without the neutralizer. It can be seen that the reduction in response with the neutralizer in operation is very substantial, and this is reflected in the long-term cumulative probability plot shown in Fig. 17.

Significant errors in tuning, i.e. $F=0{\cdot}9$ and $1{\cdot}1$, do not make a great difference to the long-term cumulative probability, and while the increase in level of vibration shown in the example is of the order of 40 per cent above optimum tuning, it is still only 25 per cent of that for the ship without the device.

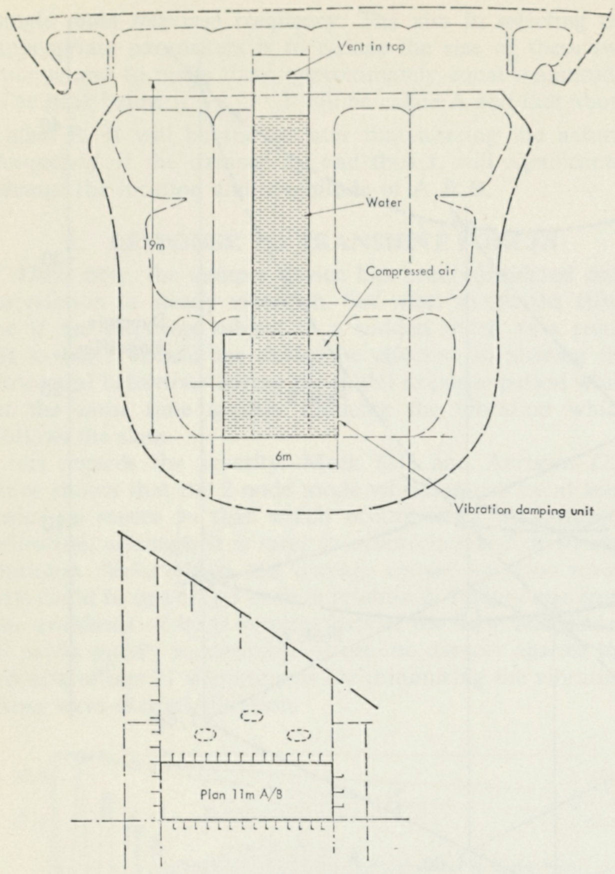

Fig. 14

Vibration Damping Unit Typical Installation.

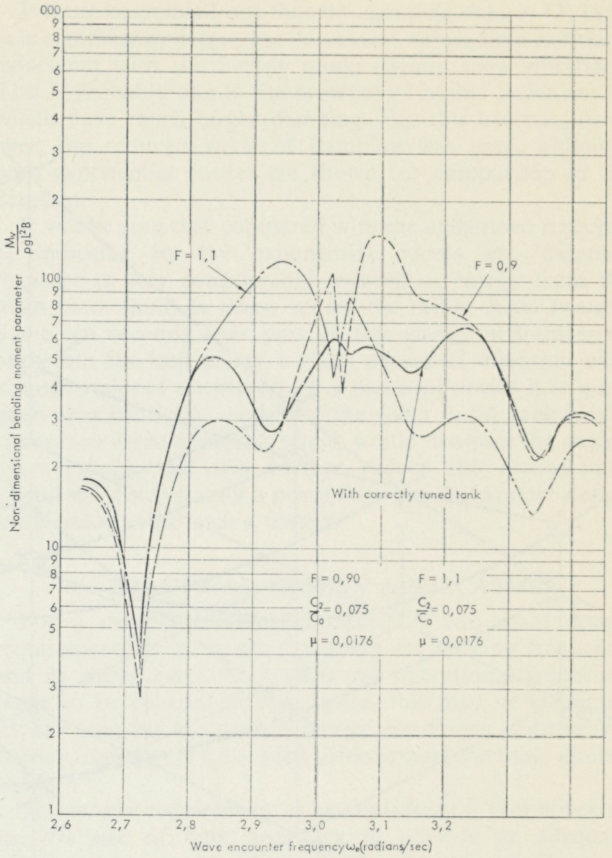

Fig. 16

Effect of Mis-Timing on Vibration Response.

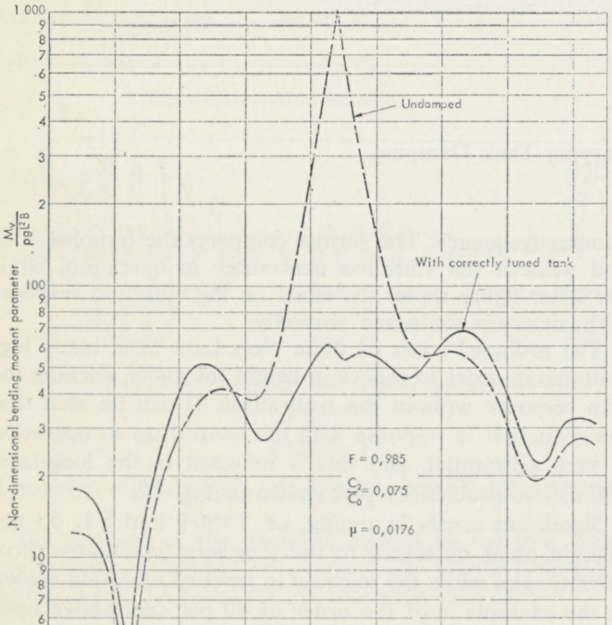

Fig. 15

Comparison of Undamped and 'Tuned' Vibration Response.

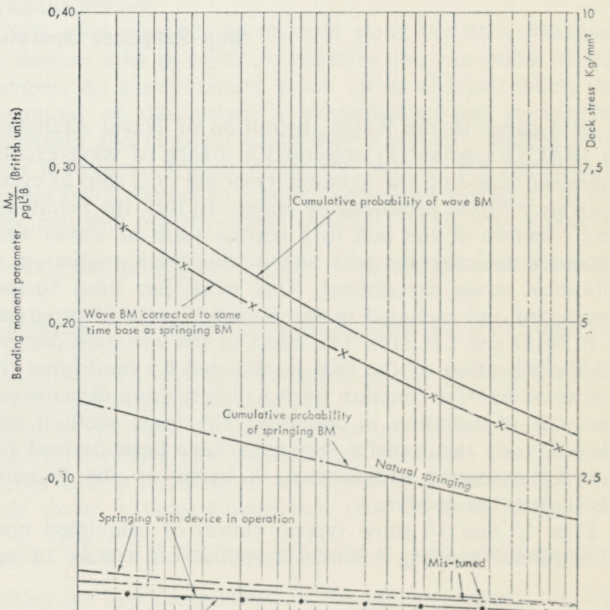

Fig. 17

Long-Term Cumulative Probability of Wave and Spring Bending Moments.

7. CONCLUDING REMARKS

The object of the final section is to gather together the main points expressed in the paper and, if possible, put forward some specific conclusions derived or expounded within the body of the paper. This task is sometimes difficult and this case is no exception. However, some pertinent facts are as follows: —

1. Springing, or more correctly the forces causing springing, are dynamic loads and, therefore, should be considered in the design process.
2. Springing causes additional fatigue damage to structural details and therefore increases the risk of structural failure.
3. Motions induced by springing can be unpleasant and tiring to ships' personnel.
4. Changes in ballast or speed or heading can affect the magnitudes of springing induced, but at the cost of loss of operational freedom.
5. A damper will significantly reduce springing, and also the transient vibration resulting from a slam.

The major factor which cannot be specified clearly is the relationship between the additional cost of incorporating such a device as a damper, and the advantages derived from a reduced springing level. Once the idea has been accepted by a shipowner, it could well become an established ship feature, just like the bulbous bow or the anti-roll device.

ACKNOWLEDGEMENTS

The Author wishes to acknowledge the assistance given to him by Mr. A. Hancock, who undertook the mathematical solution given for the transient motion of a two mass-two spring system used in Section 5. Thanks are also due to Miss P. Fleetwood for carrying out the computer calculations and the presentation of the damper design parameters (Figs. 8 and 9).

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SUBMERSIBLES AND SUBSEA COMPLETIONS

R. Hales

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SUBMERSIBLES AND SUBSEA COMPLETIONS

by R. HALES, R.N., C.ENG., M.I.MECH.E., F.I.MAR.E.

CONTENTS

SUMMARY

The paper discusses the requirements for working underwater in the offshore industry, and the various diving systems in use to the limits of the human body. Consideration is also given to observation chambers, small submarines and habitats in which personnel work at a pressure of one bar in the hostile environment of the sea.

The rising cost of fixed platforms for deep and hostile waters has increased interest in other methods of oil and gas recovery particularly from marginal fields. A review is given on the type of subsea completions which are attracting the attention of operators, to give an insight into future involvements of the Society .

1 FOREWORD

This paper has been written to introduce the various types of submersibles to show the role of such craft and the work task for which they are designed. Unlike other new fields of advancing technology, the Society has an added interest, for not only are owners required to have certification of their equipment under national regulations, for which the Society offers world-wide classification, or certification covering both the British and Norwegian sectors of the North Sea, for submersibles, but as a certifying authority under the Mineral Workings Act for Offshore Structures where owners have to utilize the services of employees at work under surveillance by offshore surveyors whilst using such equipment. A unique situation has therefore arisen.

No attempt has been made to deal with the design aspects since the basis upon which approval is given has been already published in the Rules, amendments to which are under constant development as innovations occur. It was felt that on such a wide subject a 'broad brush' review of the problems and the types of equipment would be best to stimulate interest and discussion.

2 INTRODUCTION

There is nothing new in the idea of submersibles, for it is quoted that Alexander the Great in about 332 B.C. went underwater in a glass barrel. Diving bells have been in use over the years and armoured diving suits also have been produced at various times. The advent of the SCUBA gear by Jacques Cousteau opened many people's eyes to the world underwater; however, snorkels and masks were designed by Leonardo da Vinci. The real start to the diving industry was the invention of the open and closed diving suits by Augustus Siebe in the year 1837. Developments of his ideas continued into the twentieth century to meet demands for special applications such as salvage, inshore construction work and military requirements.

Within the last decade the developments in the exploration and recovery of oil and gas from the sea bed in the Middle East, the Gulf of Mexico, the North Sea and many other waters have given impetus to innovation in diving equipment, submersibles, subsea completions, and equipment and techniques for construction, inspection, maintenance and repair underwater.

3 THE SURVEY OF FIXED STRUCTURES

On the British continental shelf, certifying authorities have a commitment under the Government regulations for the periodical inspection underwater of the various types of structures. To illustrate the time scale of the regulations the table shows the dates of some of the regulations (Table A). In the southern gas field of the North Sea there are about 50 separate tubular steel, piled structures in water depths of less than 50 m. The underwater parts of these structures were examined last year as required by the regulations. Conventional air diving equipments and self-contained underwater breathing apparatus (SCUBA) were used by the divers. The tasks and techniques involved in the surveys of these steel platforms are summarized in Fig. 1. The main objectives of the surveys were as follows: —

3.1 A general examination of each structure to determine its condition and to assess whether it was in accordance with the drawings made available by operators for the appraisal of the design. In some cases structures differed significantly from the drawings and these had to be revised and a complete reappraisal made of the design.

3.2 The degree of marine fouling had to be determined to ascertain if the thickness is in excess of the design allowance. This was found necessary as the rate of marine growth on some platforms was discovered to be greater than allowed for in the design.

3.3 The determination of the condition of the seabed, e.g., the detection of scour and the removal of debris in contact with the legs.

3.4 The state of sacrificial anodes, or where impressed current cathodic protection had been used, the condition of the anode, and associated attachments to the structure to ensure satisfactory bonding.

TABLE A

TIME SCALE OF EVENTS AND REGULATIONS COMING INTO FORCE IN UK SECTOR NORTH SEA

Effective Date	Events or Regulation	Scope
Circa 1960	First Gas Exploratory Wells Being Drilled	
1 July, 1960	SI 688 1960 The Diving Operations Special Regulations	Covering Diving under Factories Act to 3 miles Offshore and Inland
December, 1965	Sea Gem Collapsed	
1 June, 1972	S 1702 1972 The Offshore Installation Registration Regulations 1972	
March, 1974	Guidance Notes for Fixed Platforms Issued	
May, 1974	The Mineral Workings (Offshore Installations) Act, 1971	
1 January, 1975	SI 1229 1974 The Offshore Installation (Diving Operations) Regulations 1974	Covering Diving within 500 m of Fixed Installations
1 March, 1975	SI 116 1975 Merchant Shipping (Diving Operations) Regulations 1975	Covering Diving from UK Flag Ships & Non UK Flag in UK Sector
31 August, 1975	Certificate of Fitness Offshore Structures Required	
17 July, 1976	SI 940 1976 Merchant Shipping (Registration of Submersible Craft) Regulation 1976	Covering 1 Atmosphere Diving for UK Flag & Non UK in UK Sector
10 July, 1976	SI 923 1976 Submersible Pipelines (Diving Operations) Regulations 1976	Covering Diving Operations on Pipelines in the UK Sector

3.5 The detailed examination of welds particularly at structural nodes. This entailed the proper cleaning of the weld area with water jets or needle guns. Magnetic particle crack detection was used in some areas with ultrasonic thickness determination. Ultrasonic crack detection was found to be unsuitable due to the difficulty of interpretation of the results. Other methods may be used based on magnetic field but to date they are unproven.

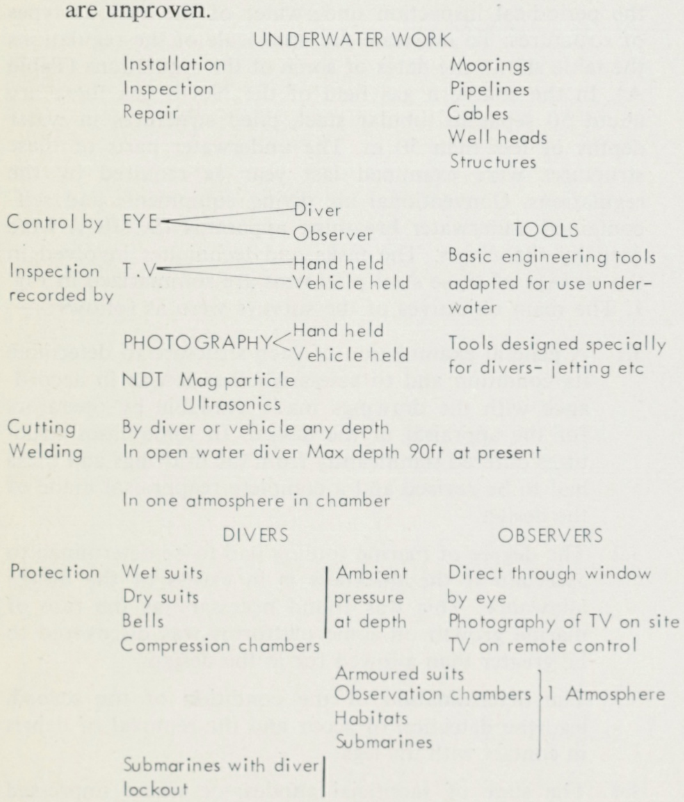


Fig. 1

Underwater Work.

In order to satisfy the requirements, these tasks have to be carried out under the surveillance of a surveyor and hence it has been necessary to use underwater closed-circuit television, video tapes and underwater photography. The work was completed satisfactorily. The main problem was the limitation on the time a diver could spend underwater because of the currents. This increased greatly the time and cost of the surveys. With increasing numbers of structures in the North Sea, it appears that diving capabilities will be stretched to the limit to cope with normal survey requirements. Clearly therefore the methods of achieving the objectives of underwater surveys have to be carefully examined.

A few structures were found to be damaged because of accident or local failure. There was inadequate time to effect full repairs. It was necessary to determine the overall safety of the structures to withstand the winter seas and to prepare for repairs in the following weather window. Welding underwater as a means of repair still presents problems despite the major advances made recently to make sound butt welds in pipelines on the seabed.

In the northern area of the North Sea, where water depths are in excess of 150 m and the structures are larger and more complex, survey and working underwater are even more complex and time consuming. Fig. 2 shows a model of one of the platforms in the Forties Field to illustrate the point. Safe access of a diver or a submersible to all parts of the structure is not possible. The trend is to make provision at the design stage for such access but this is complicated especially as the design is often amended even during the early stages of construction. The members are generally large in diameter and the total length of critical welds alone is great. These features aggravate the difficulties of cleaning and inspection. Sampling of critical areas appears to be the only reasonable solution providing reliable remote monitoring techniques are available and fitted.

In the case of a concrete structure, the surface area is so large that surveys by divers must be restricted to the more critical areas. The total surface area of a concrete structure can be over 130 000 square metres (or equivalent

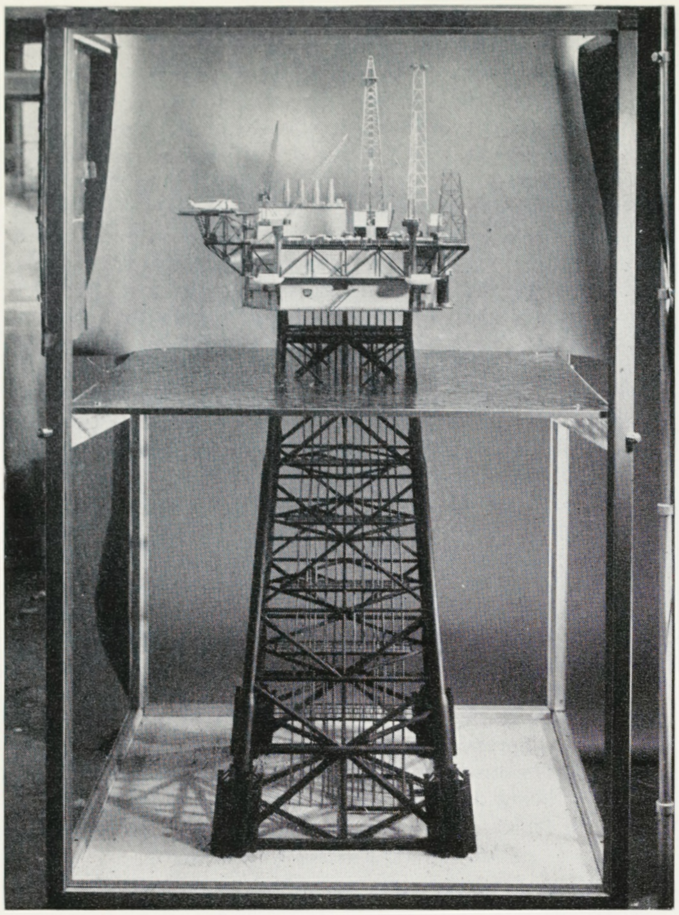


FIG. 2

B.P. Forties Field Platform Model.

to 12 rugby pitches side by side) of which only about one half is accessible. Here again, consideration must be given to the areas which require to be surveyed, access to those areas and facilities for cleaning. Clearly, remote monitoring techniques must also be considered to supplement the data obtained from actual surveys, but to date these are not yet developed.

Single point tanker loading and flare stack units give rise to similar problems of survey with the additional difficulty that the stack is free to pivot on the waves.

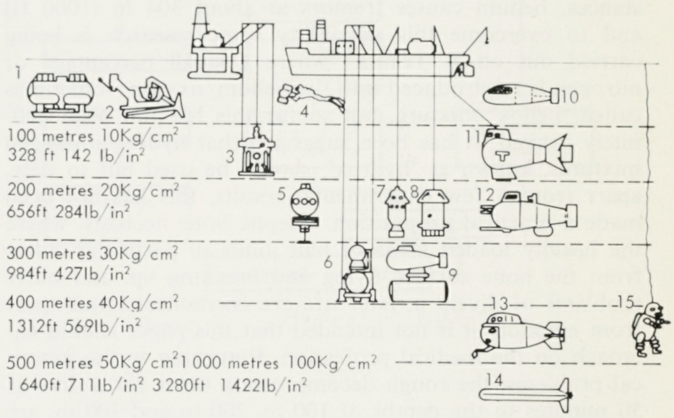
The remainder of this paper reviews underwater equipment available for surveys and subsea completions which could alleviate some of the problems in the survey of fixed structures.

4 METHODS OF ACHIEVING UNDERWATER WORK

It can be seen from Fig. 1 that a diver can be used. He may be supported by submersible equipment, or submersibles with an internal atmosphere of one bar may be used. In order to illustrate this, it is of value to examine Fig. 3 where the types of submersibles are shown and described. However, it is necessary to discuss the need for support equipment of the submersible type before we examine the details of each type.

5 WHY SUBMERSIBLES ARE NECESSARY—THE DIVING COMPLEX

Although there is nothing new in the need for submersibles, they were developed to help overcome the problems encountered by divers. These problems are also encountered by those working in compressed air, such as tunnellers for example, and are a combination of medical, physiological, mental and ergonomic difficulties as illustrated in Fig. 4. The well-known trouble of the 'bends', 'staggers', or decompression sickness has been overcome by decompressing the sufferer slowly within a special chamber at a regulated rate. These were developed from research such as that carried out in hyperbaric facilities. The principles behind air diving are widely known and the introduction of pure oxygen into a decompression cycle to reduce the time involved is not novel; however, it is not generally realized that oxygen poisoning can occur if the oxygen is at an absolute pressure of two bars, that is, equivalent to a dive depth of 10 m (33 ft).



1. Underwater habitat where men can live under saturation conditions for long periods at the environmental pressure.
2. Bottom-crawling self-propelled vehicle in this case equipped with bulldozer blade and grab.
3. Diver's lift or bell.
4. Diver with his sled.
5. Anchored observation craft.
6. Another form off subsea access chamber locked on to a subsea completion.
7. An observation bell at 1 bar.
8. Diving chamber for divers.
9. Self-propelled craft designed to lock on to undersea habitat.
10. Wet submersible for use by divers.
11. Towed manned submersible.
12. Self-propelled craft with divers or lock on access.
13. Self-propelled craft without diver's exit. Occupants remain at 1 bar pressure.
14. Remotely controlled unmanned submersible.
15. Armoured 1 bar diving suit.

FIG. 3

Types of Submersible.

	PROBLEMS OF DIVERS	
PRESSURE	Ultimate DEPTH 1800–2000ft Nervous system BONE NECROSIS etc	He/O Mixture now He O_2N_2 mixture
DECOMPRESSION	Decompression sickness BENDS	New IMPROVED SCHEDULES
COLD	External temp & heat loss by He	External HEATED SUITS Internal HEATED GASES
BREATHING	O_2 poisoning N Narcosis – CO_2 at 60M	Limit on air to 50M Mixtures below He/O_2 = Partial pressure
VISIBILITY	North Sea 2–15 ft	Maximum distance from bell limited by umbilical Lighting need navigational aids
COMMUNICATION	Donald Duck voice in He	Hand signals Speech processors
COST	Equipment needed	Time divers not employed on work

FIG. 4

Problems of Divers.

Research conducted by private companies and the various naval research laboratories has resulted in working dives being carried out to a depth of 361 m (1185 ft) and simulated dives to 610 m (2000 ft). Fig. 5 shows the problems associated with depth, and it can be seen that the development of the correct breathing gas mixtures, to maintain the atmospheric partial pressure, leads to practical problems in the design of the control equipments. Helium is introduced to prevent the dense bubbles of nitrogen building up in the body of a diver, the main cause of bends, and of nitrogen narcosis, that is symptoms of euphoria and reduced mental and physical function leading to loss of consciousness, under pressure, helium being a less-dense inert gas. There is talk that under certain circumstances, helium causes tremors at about 304 m (1000 ft) and to overcome this possibility some research is being carried out on a 'Trimix', where a small percentage of nitrogen is reintroduced into the helium/oxygen, sometimes called 'heliox' mixture, but so far this has not been definitely proved. It has been suggested that hydrogen/oxygen mixtures, known as 'hydrox', should be used but to date, apart from a few experimental results, this has not been made a practical proposition. Aseptic bone necrosis' where the heavily loaded areas of ball joints in particular suffer from the bone surface dying and breaking up, and other problems of divers in later life are encountered and apart from mention, it is not intended that this paper should encroach on the medical profession. Returning to the practical problems, the rough decompression times for a dive of 30 minutes to the depths of 100 m, 200 m and 300 m, are 3 hours, 12 hours, and 10 days, if we were able to do it, from which it can be seen that for each 30 m in a 300 m dive, the diver has to spend approximately one day decompressing, providing all goes well and the divers do not have to be recompressed because of pain and the decompression time therefore lengthened. It was the realization that the body becomes saturated with gas at pressures equivalent to depths after a given space of time that led to the use of saturation diving systems, since, once the body is saturated with gas at a given pressure one can stay saturated for a long period and still only require one decompression cycle to return to the one bar, or atmospheric pressure. This means that a diver may be kept under pressure in a surface chamber for long periods of time, in fact for days, and transported to the working site by means of diving bells without change of pressure. Also, so-called 'bounce dives' may be carried out, which means a diver living under saturated conditions at, say an equivalent of 200 m may be sent down to 300 m for work and then given a 100 m decompression cycle back to saturation pressure at 200 m in the living quarters chamber thereby saving money on the gases being used and on the final decompression time. The cost of gas is mentioned because helium is expensive and by its nature it is difficult to obtain leak tight systems. However, in the case of a deep diving system this may mean more complications in the control systems and in the number of chambers needed, so that you may have several teams of divers, each team being at various stages of decompression and/or compression. Fig. 6 shows one of many shallow types of systems being used today. Deep diving complexes are now being designed and constructed for depths of 457 m (1500 ft).

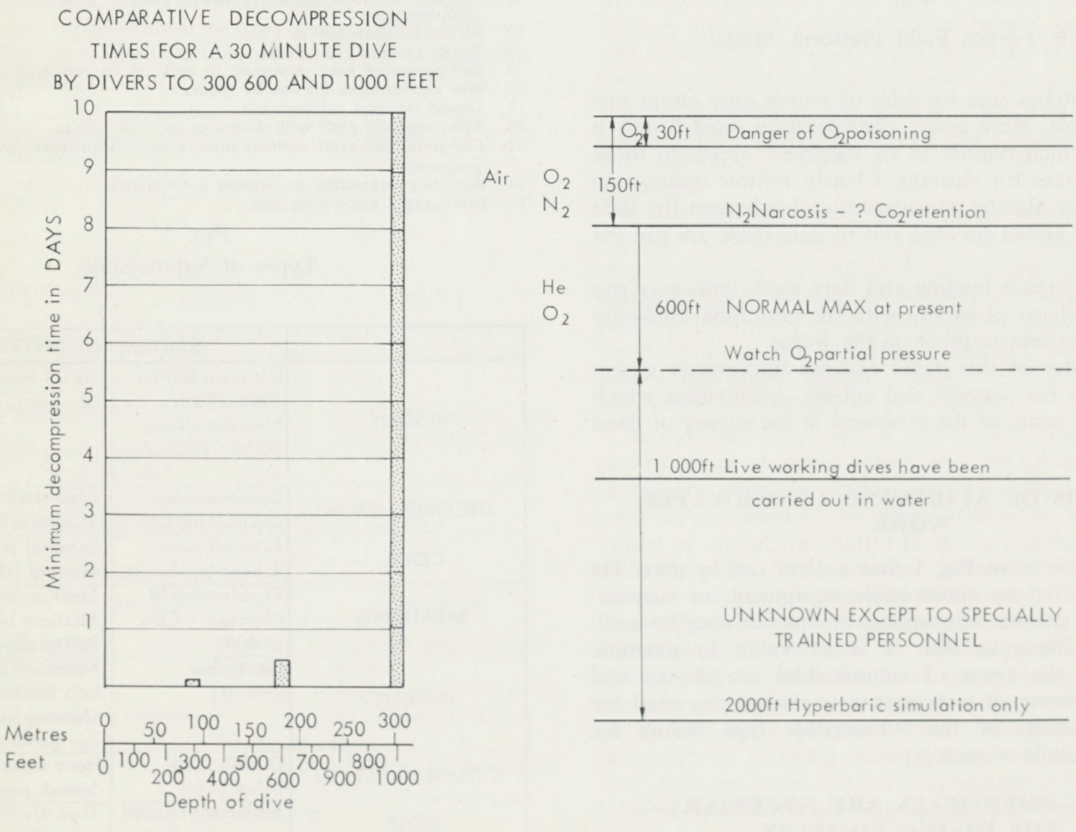


Fig. 5

Diver Depth Problems and Time.

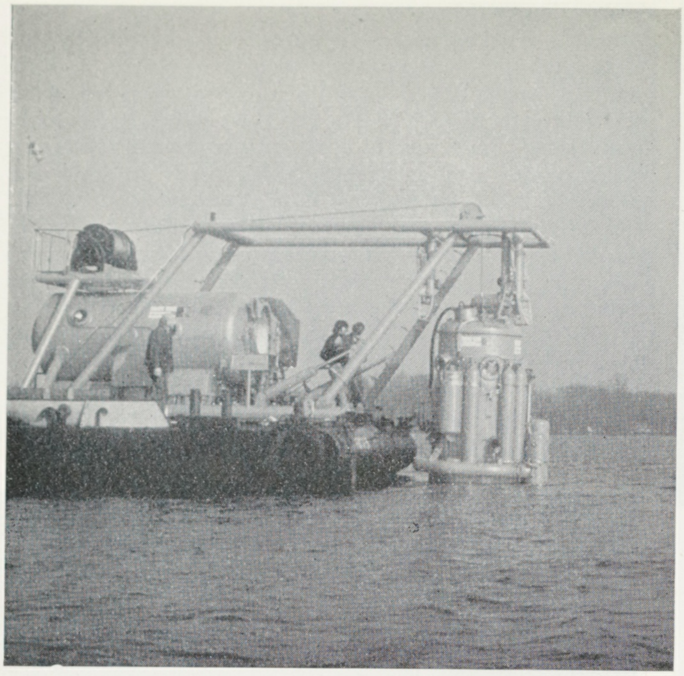


FIG. 6

Shallow Water Diving Complex.

6 OBSERVATION CHAMBER

Thus we have introduced the first type of submersible—the diving bell and the deck compression chamber. However, such a diving bell may be used as an observation chamber for one bar operation to inspect the situation underwater without recourse to the use of divers or even to determine what has to be done before sending divers down to carry out this work. This further complicates the design of such chambers, for not only are they internal pressure vessels on the surface when used as diving bells, equalizing the pressure differential before the diver emerges into the water, but on the surface, they start equalized at one bar and become external pressure vessels like a submarine, at depth.

7 BREATHING GAS CIRCUITS

A further problem exists when one is breathing a helium/oxygen mixture. The diver needs 2 litres of oxygen a minute and the breathing mixture only contains 1–2 per cent oxygen at 300 m (1000 ft) so therefore he has a vast throughput of gases through his lungs. What is called a push-pull breathing apparatus, where the gases are pumped to the diver through an umbilical and the expired gases sucked back to the bell has not yet been satisfactorily developed and the majority of the designs use open-circuit systems. To illustrate this, diagrams A and B of Fig. 7 show two types of ventilation used in submersibles. It is not within the scope of this paper to discuss divers', personal equipment since such components are in the detailed commercial field. The other factors which must be taken into account are shown in Fig. 8, of which the most important in the chambers are listed below:—

- Oxygen supply
- Carbon dioxide removal
- Contaminant control
- Temperature—since the water temperature is lower than that which the body can stand for any length of time
- Humidity
- Instrumentation—oxygen content, carbon dioxide content, temperature, pressure, trace gases, emergency gases.
- Emergency breathing

The control equipment for a diving complex is usually mounted in a control cabin which has to be manned the whole time any chamber is occupied. Fig. 9 shows such a cabin for control of the five chambers of this particular complex (Fig. 10).

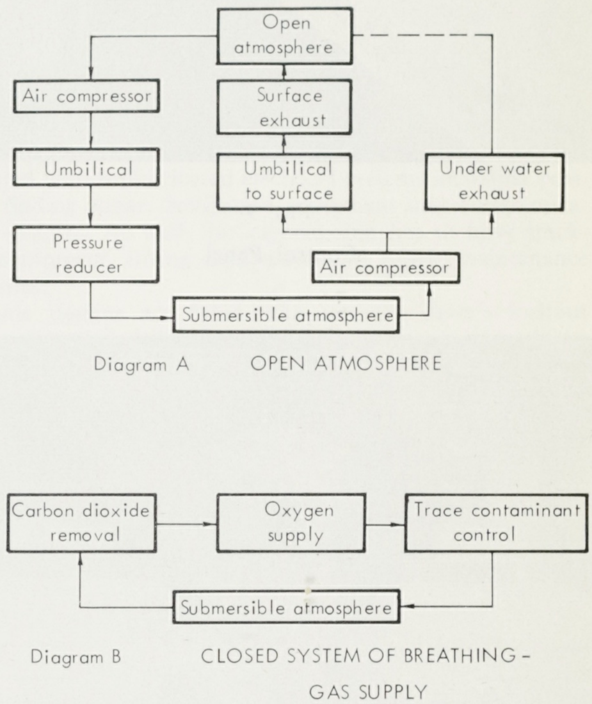


FIG. 7

Breathing Gas Circuits.

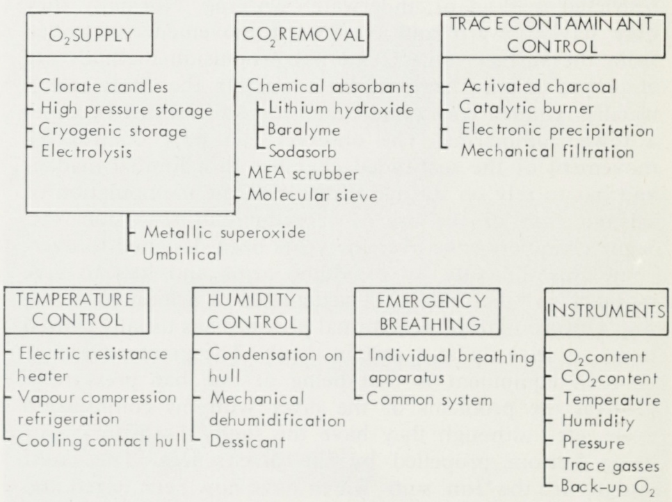


FIG. 8

Life Support Requirements.

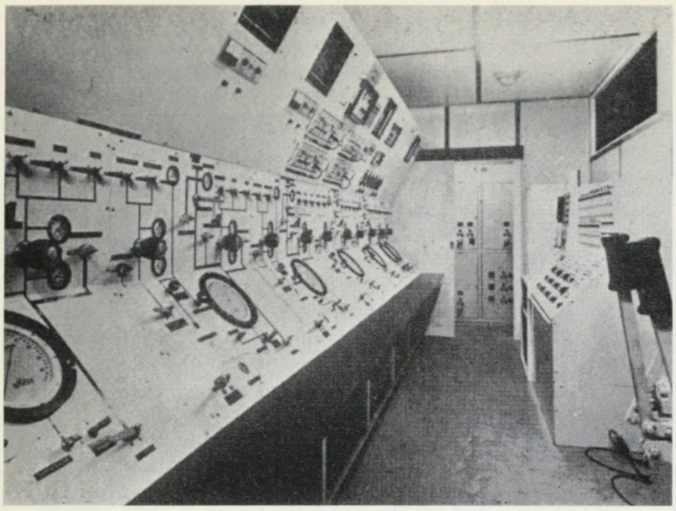


FIG. 9

Control Panel.

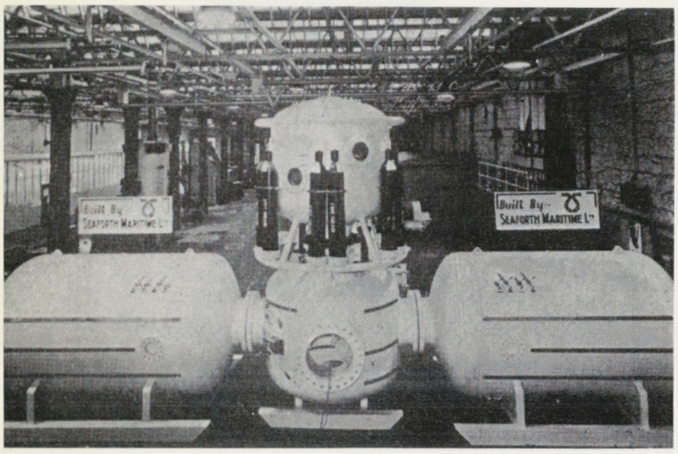


FIG. 10

Deep Diving Chamber.

As observation chambers are tethered they are a very restricted method of underwater working. Normally they only have upward and downward movement controlled from the surface, unless external propulsion methods are employed or the chambers are towed. In the latter case it usually requires control surfaces and so in effect becomes a towed submarine. The observer can only control the movement of the suspended chamber in a limited manner and has to rely on surface control for the manipulation of salvage grab or the use of closed-circuit television, etc. Some chambers have over the years been designed to overcome this difficulty by providing arms and legs to give mobility, in other words the articulated or armoured diving suit. Unfortunately, the external pressure has usually caused joints to seize up at a certain depth. The great advantage of such equipment is that being at one bar pressure it resolves the problems of the diver with his compression cycle, etc., although they have the slight disadvantage of being bottom propelled by the diver's legs. The latest designs are the 'Jim' suits, which have now been tested and are able to be used at 457 m (1500 ft), a great advance both in concept and in materials used (*see* Fig. 11). The mobility of this slightly negative buoyancy suit allows the operator to climb ladders and use his hands with the special tools to do specific jobs. The operator is also able to move forward, in water speeds of up to two knots, partly due to the fact that he is protected from the ambient pressure, whereas conventional deep divers cannot work in water speeds greater than half a knot. This ability to do actual tasks greatly improves the ideas on the observation chamber and with 'Jim' an engineer can actually go down and carry out work for as long as he likes, be raised to the surface for discussion or looking at plans and return without being subjected to decompression cycles and the need for a complete training as a diver. He does need a little experience in moving the arms and legs, but this comes in a very short space of time. However, in order to get the most out of such equipment, the subsea equipment must be designed for maintenance using such suits and with matching tools made available. These should, of course, be tried out before emplacement on the seabed.

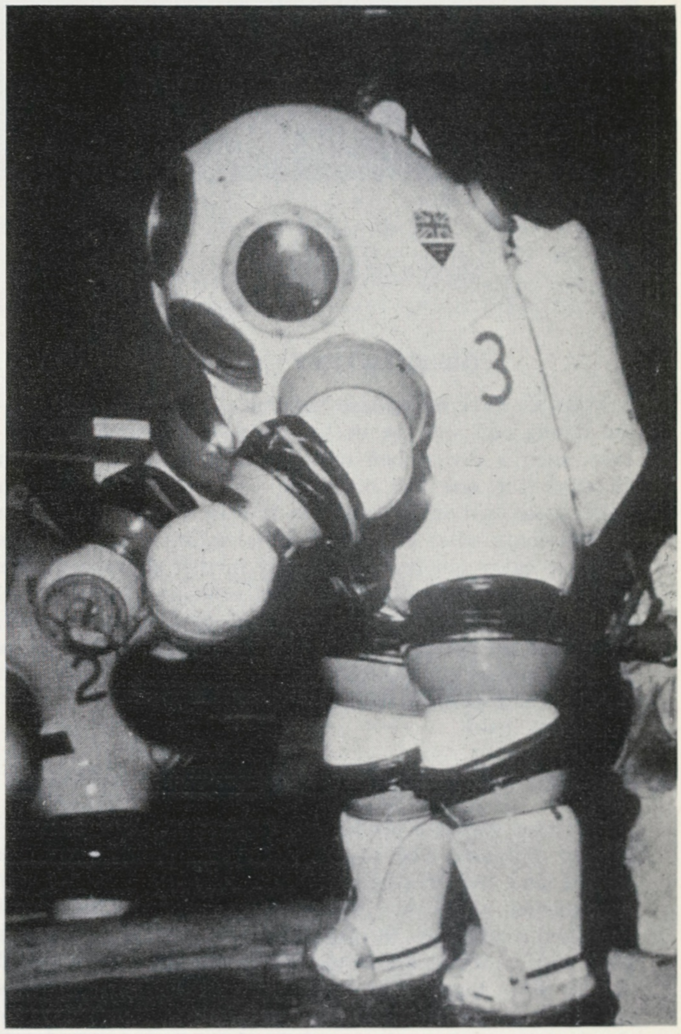


FIG. 11

Atmospheric Pressure Suit.

8 SMALL SUBMARINES

Certain tasks, such as a pipeline survey where a diver or observation chamber is extremely limited by the reliance on an umbilical from the support craft and on the short inspection time allowed by tidal conditions, bring the advantages of the completely self-contained small submarine into prominence, despite its high initial costs. They can be wet or dry, the former being illustrated by the diver transport vehicle (Fig. 12). All are limited by the amount

FIG. 12

Diver Transport Vehicle.

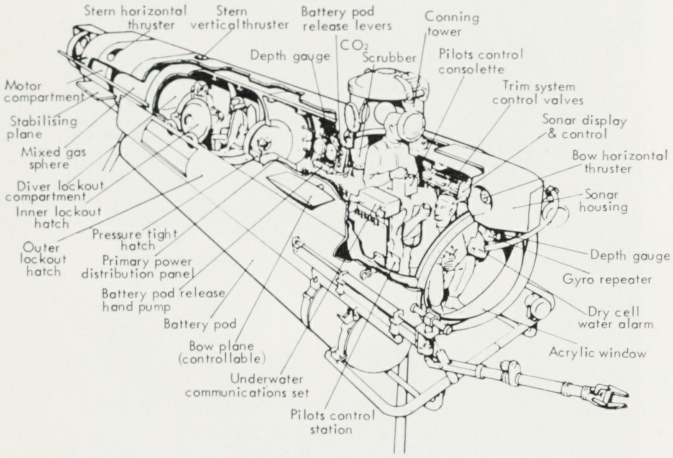


FIG. 13

Perry Boat.

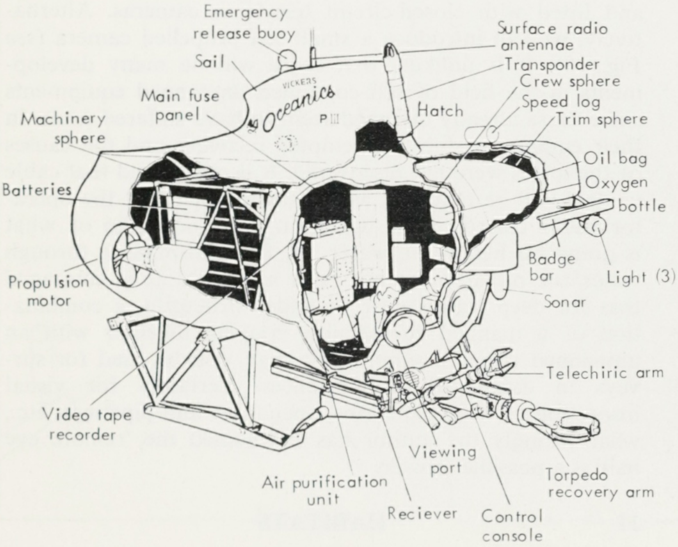


FIG. 14

Pisces Boat.

of power available within the craft and this creates problems for the designers. Due to costs and availability of other sources, standard lead-acid commercial batteries are usually used in pressurized containers, and fitted with suitable electrical protective devices and arrangements for degassing whilst being charged. It is significant that from original exploration submarine designs of the 1960's which were one-off, very high cost craft designed for specific purposes, only two or three commercial designs have developed. In the first place, these have not had the deep-diving capability of the original craft and have been designed to operate within the confines of the continental shelf area. Therefore, they are usually designed to operate in a maximum depth of between 450–960 m (1500–3000 ft). The two well-known designs are shown in Figs. 13 and 14. Both these craft are entirely self contained, having the same requirement for the life support as the observation chambers but for longer periods. They are fitted with propulsion, means of changing buoyancy, trim, and steering gear, coupled with sophisticated electrical systems including position finding sonar, navigation equipment and communication systems. As well, the support ship has to have tracking equipment, lifting equipment and repair/maintenance facilities.

Some designs are fitted with separate diver's lockout compartment and thus combine the advantages of both the submarine and the diving bell. In such cases, the support ship is fitted with a compression chamber to which the diver can transfer when the submarine is locked on to it for completion of his decompression cycles.

The same designs can also be used in the same manner as an observation chamber in order to transfer workers at a one bar atmosphere to a one bar chamber underwater in the dry condition, as demonstrated by the submarine shown in Fig. 15.

9 LIFTING ARRANGEMENTS

A common problem of all submersibles, especially in the severe marine conditions of the North Sea, is the question of lifting appliances, frames, cranes, winches, brakes, wire

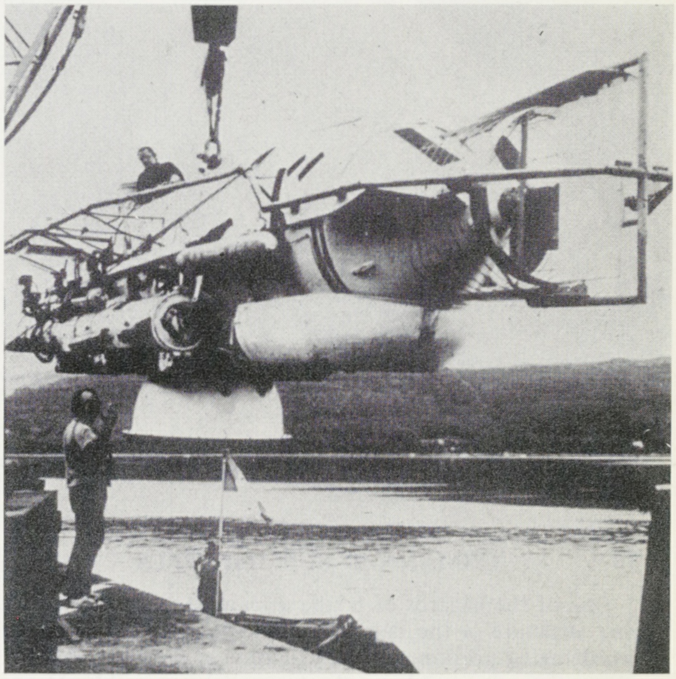


FIG. 15

Submarine with Dry Transfer Skirt Attached.

and umbilicals, etc. The type which is to be employed depends on whether the submersible is being operated from a ship, a platform or a semi-submersible craft. In the case of the ship, some have been converted to take an A-frame over the stern and working with constant tension winches are able to lift the submarine from position downwind or downsea on to the deck. For diving bells, certain ships have even been equipped with moonpools fitted in such a position where the ship motion is a minimum both fore, aft and athwartships. Even so, new problems are met and it is now clear that a bell returning into a moonpool must be aligned and be held before it enters the tube through the ship's hull to prevent damage to the bell and to the occupants. A typical A-frame is shown in the photograph (Fig. 16).

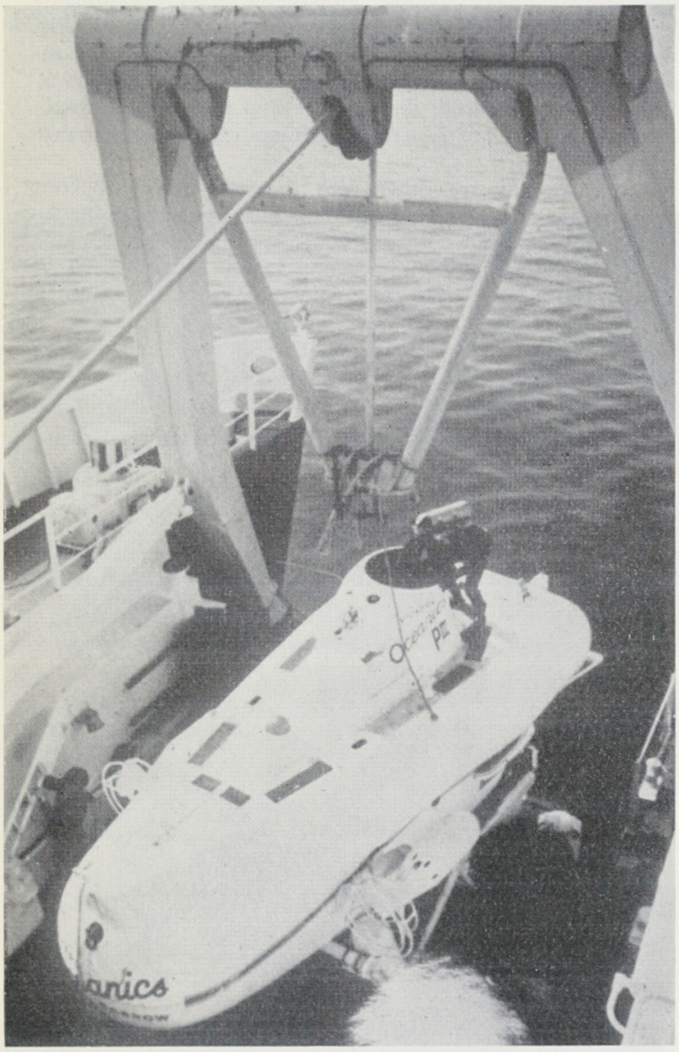


FIG. 16

Pisces and A Frame.

10 UNMANNED SUBMERSIBLES

One of the hazards of taking a manned submersible into any structure is the danger that it may be snared by an overhanging portion of that structure, making recovery of either the submersible or the personnel impossible. Hence the need, in certain cases, to use unmanned surface or sub-surface controlled equipment and these are illustrated by introducing the BAC Consub (Fig. 17), which is tethered

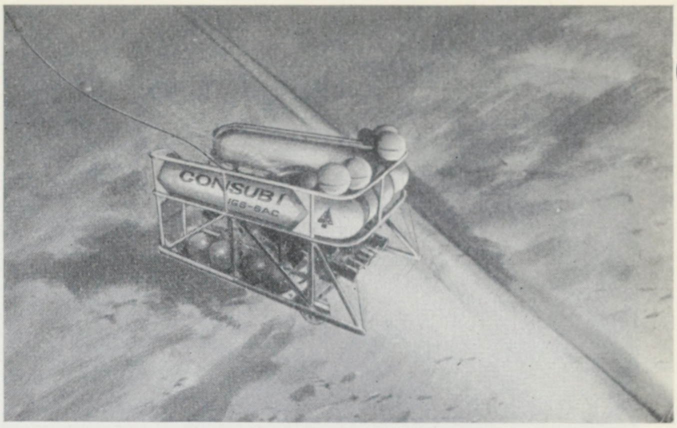


FIG. 17

Unmanned Tethered Craft.

FIG. 18

Unmanned Tethered Mini Television Craft.

and fitted with closed-circuit television cameras. Alternatively, we can introduce a small self-propelled camera (*see* Fig. 18). It is unlikely that there will be many developments in the field of self-contained unmanned equipments since those already designed seem to have suffered faults in their equipment during attempted recovery and the causes of the losses were not discovered. It is considered that cable control will be required, as well as the fact that the operator needs closed-circuit television to inform him of what is going on below the water. So far transmission through water for pictures has not been a success. It is envisaged that for deep waters near or under structures, a combination of a manned submersible coupled together with an unmanned tethered submersible may well be used for surveys in dangerous circumstances. Certainly for visual inspection of deep platforms, anchorage and pipelines, etc., what jokingly the author has nicknamed the 'remote eye ball' is a possible answer.

11 HABITATS

Since underwater habitats were developed, that is submersible living quarters where divers lived under saturation conditions for several days, changed into wet suits and

emerged for specific biological investigations from time to time, no experimental work for commercial applications has been made. However, due to the problems associated with developing wet-welding techniques, welding habitats have been used in commercial applications. One such habitat has been constructed as a pipeline welding chamber and pipe alignment frame (shown in Fig. 19). This has been successfully used in many applications. Another type is a riser connection welding chamber (Fig. 20). These have been built for the Thistle Field. It now appears that the atmospheric pressure chamber will become more common in the subsea completions, where the transfer at a presssure of one bar will be used, in order to overcome the diving problem.

12 SUBSEA COMPLETIONS

By 1975, some 252 so-called subsea completions were listed in the technical press but only 26 are true subsea completions. These may be described as being the 'completion of producing well in which the producing Christmas Tree and all other primary well controls either exposed to the water or fully encapsulated are located on the ocean floor'. The Christmas Tree is the name given to the blow-out preventor stack mounted on the casing through which drilling or production of oil is controlled. Actually, there are two phases involved in this case. Phase I is the drilling stage and Phase II the production stage. In the land petroleum techniques the drilling is carried out through blow-out preventors and the well is controlled by mud pressure, the valves of which are all placed on the surface of the ground; but when the drilling is some hundreds of feet above the ocean floor and the drilling is through the seabed, long lengths of casing carrying the mud have to be attached to the seabed through which drilling is occurring. If the blow-out preventor was situated at the drilling platform then these long lengths of pipes are a potential hazard and pollution of the sea surface is inevitable should leakage or even a fracture occur. Also in bad weather, when the strings of casings and drills etc. have to be disconnected, leakage would occur. Hence the need for two sets of valves, one on the ocean floor or in the down hole and the other on the platform. Likewise, once the production phase is reached then the risers become the hazard and the initial control must be located as near the source as possible. With a few exceptions, most of the subsea Christmas Trees and

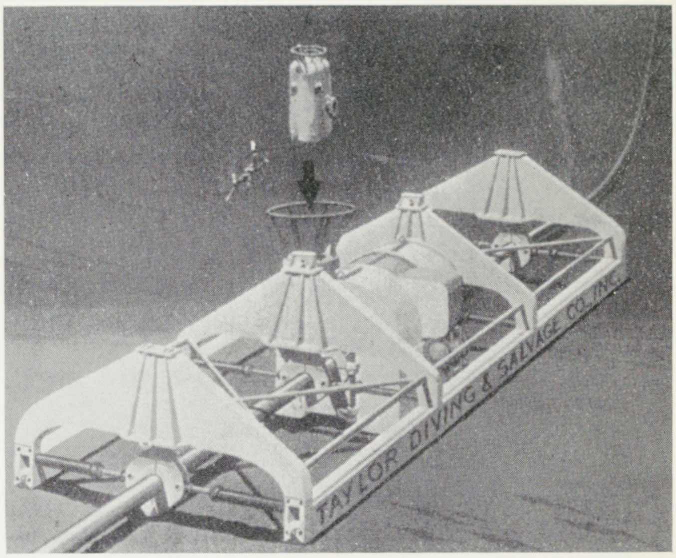


FIG. 19

Pipeline Alignment Frame and Welding Chamber.

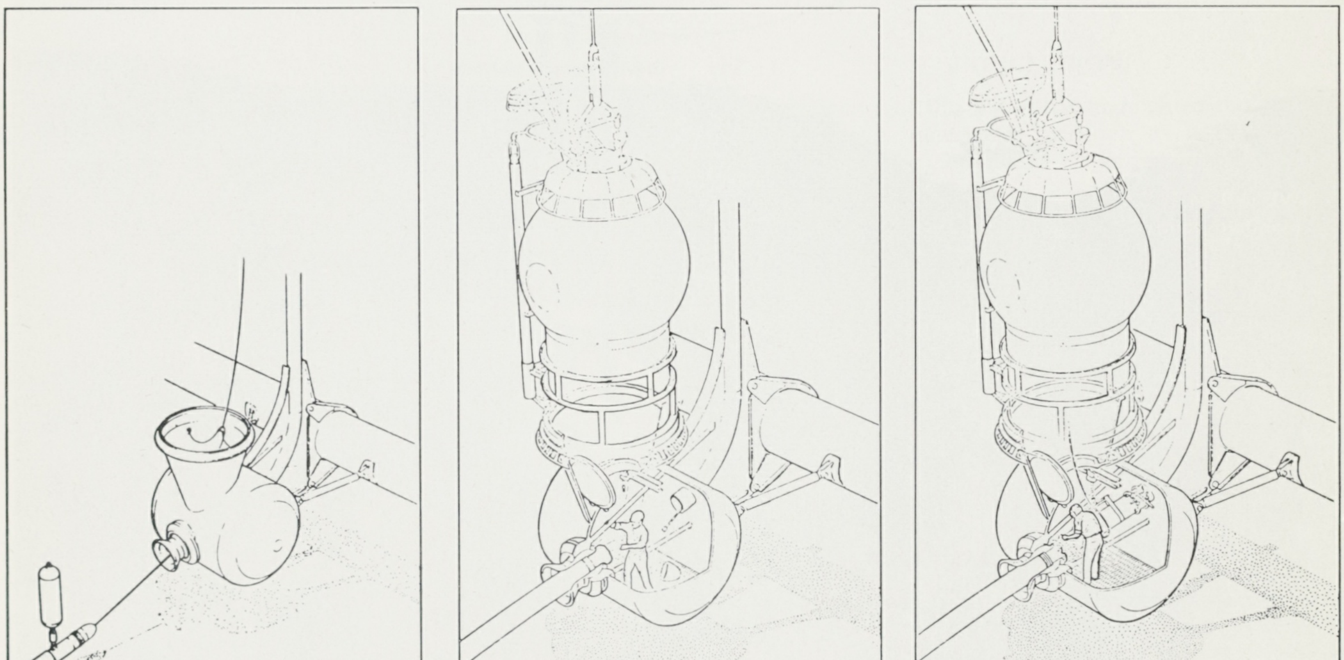


FIG. 20

Riser Connection Welding Chamber.

blow-out preventor stacks have been designed for operation remotely from the platforms and diver intervention for breakdowns of the equipment has been found to be necessary.

Three types of system have been evolved:—

1. A framework which remains in place for a long period of time called a template, and one design is illustrated in Fig. 21, in which mechanical operators are run on rails within a large framework and the drilling operation is carried out through part of the frame and the well-heads are all brought together, the whole of which is covered by inverted drip pans as a safety system for protection.

2. The subsea Christmas Tree is remotely operated either by electrics or hydraulics; and examples are given in Fig. 22. Up to now they have found that divers may be required under certain circumstances although they can be designed for complete remote control and tested remotely.

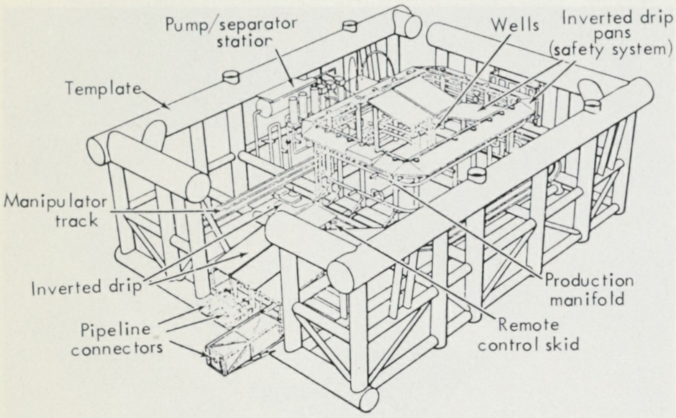


FIG. 21

Template Subsea Completion.

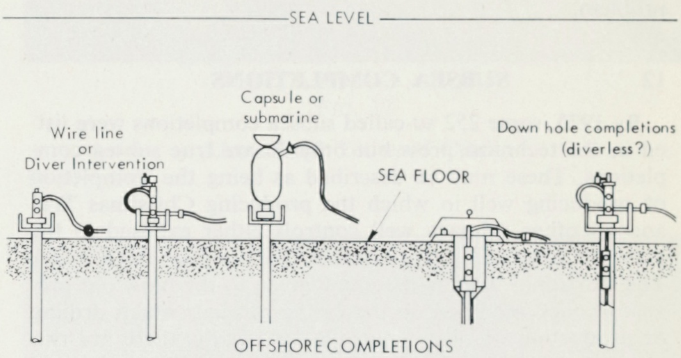


FIG. 22

Subsea Completion Single Types.

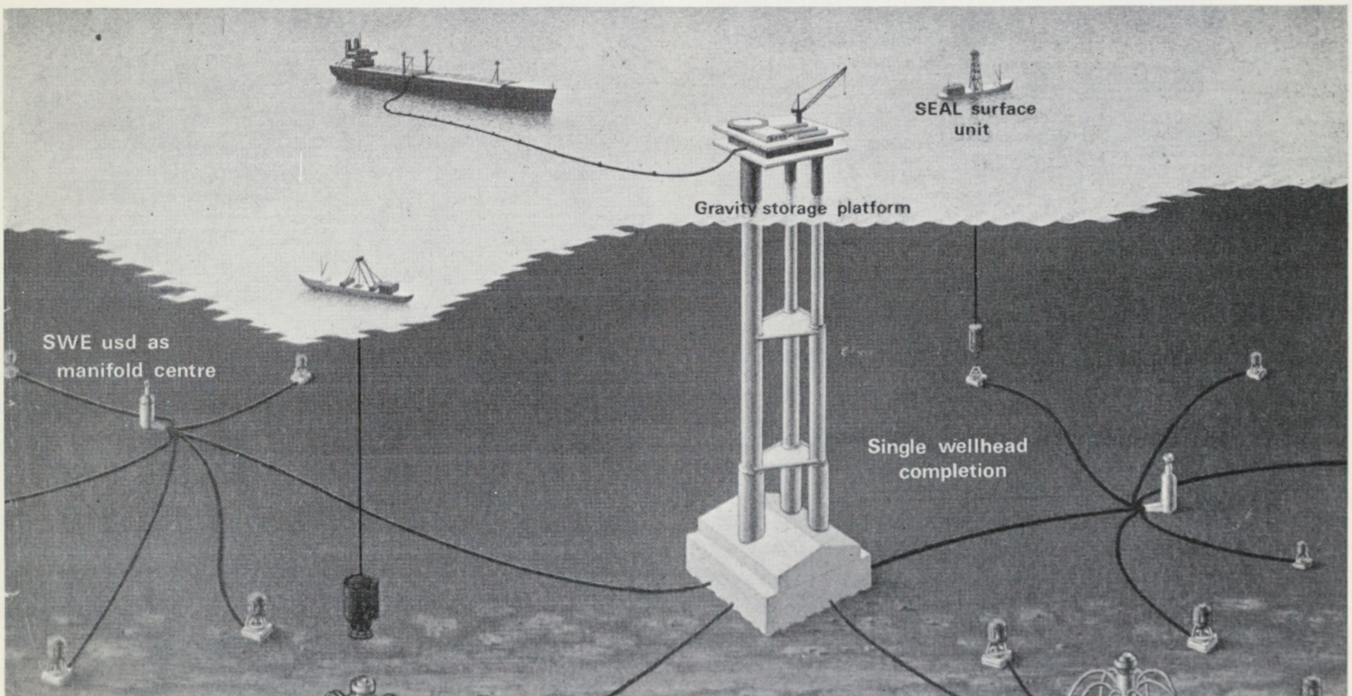


FIG. 23

Multi Subsea Completion.

3. The completion is designed for replacement and maintenance once installed underwater. All vital valves etc. have been placed within a one atmosphere chamber which is capable of being mated either to a bell-type capsule, an observation chamber, or to a submarine so that maintenance staff can descend, enter and work inside at a pressure of one bar. There are two ways of achieving the answer: —

   (a) Using a wet chamber the men go from the one atmosphere observation chamber in diving equipment but only dive at a pressure of one bar. The maintenance staff wear breathing apparatus and so dangerous gas concentrations are eliminated.

   (b) or using a dry chamber where the staff have a working shirt-sleeve environment. To illustrate this type of design and layout see Fig. 23.

To illustrate the importance of such developments and to look into the future it is worth considering the way that costs of fixed platforms have escalated from £50 million at 100 m to £200 million at 182 m whilst the production equipment has only increased from £90 million to £100 million. I have not taken into account semi-submersible designs, single point moorings etc. which will be required for subsea completions in North Sea. Coupled with this is the fact that the well pressures will be averaging at about 278 bars, that is 4000 lb per square inch. Any spillage from a fractured pipe will contaminate a large area of the sea before it can be shut off. If any failure of valves occurred, when fitted on a platform, the result would be catastrophic and therefore the subsea completion where maintenence on the sea bed can be carried out becomes a vital consideration.

With maintenance in view, any one bar chamber has to be tested and inspected at regular intervals before being re-entered either from a submarine or an observation chamber and these tests have to be conducted as the vehicle is coupled to the chamber. This introduces the fire risk and toxic hazards of the petroleum production, that is, the vapour fractions of gases such as methane, hydrogen sulphide etc., which have to be kept to low percentages if the men are to work satisfactorily without having to wear breathing apparatus. This stresses the importance of the certification of these chambers and of the systems involved in subsea completions, and discussions on the requirements are being initiated with the authorities.

It is envisaged that there are four stages of the subsea completion which have to be considered for certification, these are: —

STAGE 1 — The manufacture of all items under survey etc.

STAGE 2 — The emplacement on the seabed.

STAGE 3 — The connection to the oil production equipment.

STAGE 4 — The in situ subsequent inspection of the habitat and production equipment.

STAGE 1 (a) The chamber must be constructed to the requirements of the certifying authority and from materials suitable for use at a reference temperature of $-10°C$ with a suitable corrosion allowance.

(b) Internal equipment should be designed to the appropriate code of practice.

(c) The access chamber also must be constructed to the requirements of the certifying authority.

(d) The life-support gas and electrical systems should be built in accordance with the certifying authority requirements, remembering that an enriched oxygen atmosphere may exist. Sufficient life support should allow for a seven-day capacity per man.

(e) The seabed mounting must be such that external parts are faired to prevent snagging yet allowing for access for external examination.

(f) The buoyancy chamber must be such that it can be taken to site and under control lowered to the seabed to a fixed location.

(g) Internal equipments should be designed so that periodic examination and refit can be made on the surface.

STAGE 2 (a) The seabed conditions, i.e. current subsea state, likelihood of scour, must be determined.

(b) Diver or remote television will be required after emplacement to check position and anchoring devices.

(c) Internal inspection is required.

STAGE 3 (a) Access for inspection. Access capsule to be able to monitor the habitat before opening the hatch including atmosphere and fire hazard.

(b) Thermal expansion problems between habitat and production equipment must be considered.

STAGE 4 Routine inspection at given intervals with access as for STAGE 3 then detailed testing will be required of the chamber and of the systems.

13 CONCLUSION

It is hoped that this paper has highlighted the problems associated with underwater work or, as it is sometimes called, hydrospace. The surge of activity created by the economics of fuel recovery has caused the advance of technology at a rate which has never been equalled before, except perhaps in the aerospace industry with which there is a close affinity, although the relationships of pressure are reversed and of lower values.

It was not until 1956 that a man reached a depth of 200 m (600 ft) in a diving suit and 1960 for a manned observation chamber to reach 10 942 m (35 900 ft). Yet in the last eight years the necessary knowledge for a diver to descend from the previous limit of 200 m has been increased so that working dives of 600 m are a possibility.

14 ACKNOWLEDGEMENTS

The author wishes to express his thanks to Mr. G. P. Smedley for his help and encouragement and to those colleagues in the Society, the industry and in the British and Norwegian Diving Inspectorates who are assisting in the formulation of the agreed national requirement and Society's Rules to cover the equipments in this rapidly expanding field of technology. He also wishes to thank Mr. G. Pumphrey for producing the illustrations and typists in the Offshore Services Group who have typed the various drafts.

The author also wishes to gratefully acknowledge the assistance of the following companies in permitting the use of illustrations as follows:—

B.P. Ltd. (Fig. 2)
Barry Henry Cook Ltd. (Figs. 9 and 10)
D.H.B. Ltd. (Fig. 11)
Sperry (Fig. 12)
Perry Oceanographics Ltd. (Fig. 13)
Vickers Oceanics Ltd. (Figs. 14, 15 and 16)
B.A.C. (Fig. 17)
Hydro Products (Fig. 18)
Taylor Diving (Fig. 19)
Lockheed Petroleum Services Ltd. (Fig. 20)
Exxon (Fig. 21)
Seal Petroleum (Fig. 23)

15 REFERENCES OF INTEREST

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16 LIST OF STATUTORY REQUIREMENTS APPLICABLE

1. The Merchant Shipping Act 1974, Chapter 43 Part IV Sections 16 and 17.
2. Statutory Instrument No. 116 1975, The Merchant Shipping (Diving Operations) Regulations 1975.
3. Statutory Instrument No. 940 1976, The Merchant Shipping (Registration of Submersible Craft) Regulation 1976.
4. The Mineral Workings (Offshore Installations) Act 1971 Chapter 61.
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7. Statutory Instrument No. 923 1976. The Submarine Pipelines (Diving Operations) Regulations 1976.

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Discussion

on

Mr. R. Hales's Paper

SUBMERSIBLES AND SUBSEA COMPLETIONS

Paper No. 1. Session 1976-77

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Discussion on Mr. R. Hales's Paper

SUBMERSIBLES AND SUBSEA COMPLETIONS

Mr. R. N. YEABSLEY

I found the paper extremely enlightening on such a vast subject, but some of the terminology used was rather foreign to my experiences, which leads me to ask the question, what is a moonpool?

Mr. M. H. P. HEMBLING

My congratulations to the Author on a very informative paper which has given us all an insight into some of the many varied problems unique to the offshore industry.

The Author spoke of a temperature difference of only 4°F between hypothermia and hyperthermia for a diver at a depth of 1000 ft.

Does this refer to the body surface or core temperature and could he explain further the methods used to keep it within these limits?

Mr. J. E. STEVENS

The reluctance on the part of Mr. Hales to linger long on medical and physiological aspects of his subject is appreciated. The task, well achieved, of presenting the technical information alone is daunting. I would, however, request the author to comment on the physical protection offered now, and envisaged in the future, to counter underwater noise.

Personal experience of this noise up to depths of 50 m have impressed the non-directional and irritating aspects on me and I understand at greater depths, with heavy surface traffic, the condition becomes intolerable.

Mr. E. L. GREEN

Like many other Surveyors in the Society who have no personal experience of this fascinating new technology, it could possibly be useful to bring to bear on this subject, a mind uncluttered by any fore-ordained or circumscribed views and indeed unhampered by any special knowledge of the parameters involved.

Could I therefore raise a few queries (not too outrageous, I hope!) and make some observations regarding the survey of fixed structures.

1. It seems to me that the position with regard to the survey of these structures is somewhat analogous to that of nuclear pressure vessels in that there is being put into service a piece of equipment which we shall never again be able to inspect in a conventional manner. Superior manufacturing techniques are therefore required and arrangements should be incorporated in the design for a built-in inspection system with external recording.
2. Considering the fearsome cost in time and money (not to mention the risk to life and limb) for the underwater inspection as described in the paper, would it not be possible to see how much inspection could be carried out from the inside?
3. Bearing in mind the extensive armoury of inspection and measuring devices including, ultra-sonics, radiography, television cameras, strain gauges, telemetry etc., surely means could be provided inside the structure whereby thickness determination, subsidence, distortion or any other physical change could be measured and recorded. In para 3.5 the Author refers to examination of the welds by magnetic particle detection and ultrasonic determination. The inference here is that this is carried out from the water side, is this correct?
4. With regard to marine fouling and scouring, could not use be made of test modules (perhaps of quite substantial size) which could be lowered to various depths adjacent to the structure and hauled up at intervals for examination and assessment of their condition.
5. By its nature, underwater examination even under ideal conditions must be fraught not only with the many hazards peculiar to its environment, but also some absence of definition and I put it to the Author that a scientifically conceived internal inspection (although by no means replacing an underwater examination) would be a formidable complementary tool.
6. From the photographs I have seen of various fixed structures, it may be that no meaningful internal examination was ever intended and if so it is probably too late to do much about it now and, as in the case of nuclear vessels, improved in-service inspection techniques could only be applied to a future era of structures.

AUTHOR'S REPLY

Mr. R. N. Yeabsley

A moonpool is a trunking extending down through the hull of a ship, pontoon or semi submersible from the deck above the waterline through which equipment, i.e. drill-strings, casings, B.O.P. stacks etc., and diving bells can be lowered. Usually they are sited over the longitudinal centre of buoyancy and amid ships so that the effects of the ship motion are lessened. Although "slosh" does create problems, involving in the case of bells special "slosh" splitters and hatches. In some cases it is necessary to catch the bell in a frame before raising/lowering it through the trunking. Also attempts are being made to change the density of the water in the moonpool in order to alter the frequency of the motion, by using a perforated pipe and blowing air bubbles up the sides of the pool.

Mr. M. H. P. Hembling

The question is an extremely difficult and complex one to answer since both are involved. Once one moves into the field of dry suits, as opposed to wet suits where the core temperature only is controllable by using heated gases or the water flow over the body surface is supplemented by a hot water flow bleeding out at the neck, hands and feet of the diver. In dry suits we are talking of divers working in areas where the air temperature is anything from −60°C to 60°C, but once below usually the temperature does not change very much from 4°C to 8°C. However, the problem which arises is that the core temperature is affected by the surface heat exchange rate especially in a helium atmosphere where the heat transfer rate is treble that of the air. Research has shown that the parts of the body especially the extremities e.g., legs, upper and lower, arms, upper and lower, and the trunk upper and lower are affected at different rates. The core of the upper trunk for instance is rapidly affected by the breathing gases, as compared to the

lower trunk where there are few major blood vessels actually near the surface as in the lower arms and legs. In order to overcome the surface temperature hot water heated suits and some electrical suits have been used, also hot/cold air suits of a blow-up or inflatable type are available. The last mentioned is useful in both the tropics and in the Arctic. Electrical suits or heated underwear are available, but not really satisfactory and more development is taking place, since wires embedded in the material make movement difficult and there is always the possibility of fracture. They are therefore potentially hazardous to the diver (cf. the electric blanket at home). However, the fact that the diver only has wires to be manoeuvred with his breathing hoses would be advantages compared to his lack of manoeuvrability with hot water hoses. Hot water suits have small tubes running round the section of the body and thus areas requiring more heat can easily be controlled by the number, unlike the electrical suit which needs separate circuits to each part of the body.

The upper trunk core, where the heat transfer in the lungs is the problem, it is usual to heat the gases for breathing with either hot water or by electricity.

The 4°F quoted is that found by experiment on the healthy average person in a helium atmosphere and merely shows that in order to be safe one must consider the problem in that light and the diving supervisor has to do his best to control the situation.

Mr. J. E. Stevens

In reply regarding noise underwater it is agreed that noise is a problem and at present the only answer is to try and reduce the effect by the use of a helmet and earphones suitable for the actual diver. It is agreed that this is no answer, but is difficult to envisage being able to *stop all machinery* within say 2 miles of a diver underwater.

Other water noise problems exist for those in one atmosphere submersibles—internal noise echoes around becomes intolerable at times, whilst the noises produced by fish—dolphins, whales etc., affect sonar operations.

Mr. E. L. Green

It is assumed that Mr. Green is unaware that all the original rigs designed, built and put into commission were designed without consideration for on site inspection. No access was designed into the structure, and legs are used as tanks, the struts are welded onto the legs and there are no cutouts for stress reasons. It is difficult to imagine a person, let alone a surveyor climbing down inside a tube 300-400 feet below the waterline with no ladders or safety rails. Scientific investigations and research are being carried out on the possible use of sonic signatures, but so far it appears that the complex nature of the structures is such that a large computer has to be installed very early in the life of a rig, with many sensors and much wiring, in order to establish the base signature. This makes it extremely costly and even then no standards of acceptability have yet been established. Also the effect of running machinery will mean that a complete shut down is required each time a new reading is taken to match the base signature, however research is being carried out on some platforms to pick up the vibration pattern under all conditions on a continuing basis.

The use of strain gauges is current in some areas, but not usually underwater since means of sealing the gauges, and wires in seawater are unreliable and therefore of only limited use at present. Automatic tracked and guided cameras have not been too successful, and will only be used if one particular area needs constant inspection.

Thus we are left with the simplest possible means of using ultrasonics, as rightly guessed, from the sea water side for thickness determination and visual inspection backed up by magnetic particle testing where necessary, the divers being guided by the surveyor on the surface using underwater closed circuit television; or the use of the "remote eyeball" which is proving a very useful tool. Here again problems concerning the identity and position of the area on such large structures create difficulty, and each operation needs careful planning. Operators need training and it is understood that few can successfully navigate the eyeball for more than a few hours. Having viewed the screens myself I can vouch that it is difficult to keep track of where one is looking even with a voice commentary.

It is agreed that if the structures had been designed with inspection in view then the scientific research to find a means of achieving satisfactory alternatives to manned intervention could have been initiated earlier. But it should be remembered that the economic situation created a time scale such that the companies who had the leases in the North Sea blocks had no time for studies and practical implementation of any schemes devised. Their need was to obtain oil. It is only now as new designs are being developed, using the hindsight of previous experience and of human limitations, that these technical details are being considered together with the economics covering the need for such equipments.

In reply to the suggestion of using test models to determine rate of fouling, it should be noted that the majority of the fouling is occurring in the inwater light zone down to about 180 ft. from the surface, inside easier diving depths. Below that, the fouling depends on the attachment of hard encrustations and if one sends down test pieces these have to be secured to the structure, divers would be needed, an unnecessary expense when you have the structure there anyway. It is also useful to know that once one of these structures is emplaced it becomes a "reef" upon which marine life flourishes and a small object does not attract the plankton in the same manner; the plankton attracts the fish and invertebrates etc. This was only found out by experience as the original trials on small objects showed little fouling at depth.

Scour refers to the movement of the surface of the seabed and it cannot be reproduced without a full scale test object. Underwater currents resulting from storms, tides and their seasonal variation over the whole area of the continental shelf together with the flow pattern of the water round structures and pipes and the nature of the soil, its type, size of particles and compaction, all contribute to the migration of soil in the vicinity of any bottom structure. Scour mats and skirts are usually placed around the structures where it has been found that scour is likely to occur and they have been reasonably successful.

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CARGO OIL PUMP INSTALLATIONS

Some Aspects of Design and Operation, and Problems Encountered

K. M. B. Donald

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CARGO OIL PUMP INSTALLATIONS

Some Aspects of Design and Operation, and Problems Encountered

by K. M. B. DONALD

Synopsis

The purpose of this paper is to put on record some aspects of design and operation of vertically and horizontally arranged cargo oil pump and associated machinery installations which were given consideration as a result of problems encountered in the early 1970's, for which the Society's Research and Technical Advisory Services were requested by Owners, Shipyards, and Manufacturers.

The contents are not a treatise on the specialised subject of Pumping and Piping, but are intended rather to reflect the range and types of problem dealt with in an advisory capacity and applying the principles of good engineering practice.

INTRODUCTION

Liquid cargo pumps and the associated driving and transmission units installed in tankers play an essential role in trading throughout the world.

Crude oil carriers have always been equipped with their own pumping units almost solely for the purpose of discharging their cargo at oil terminals on completion of a voyage.

Products, chemicals, and L.N.G. tankers are equipped with cargo handling pumps to similar design, but this paper deals mainly with the crude oil carrier cargo pump installation, and some of the problems encountered by the Research and Technical Advisory Services Department.

It has been estimated (Ref. 1) that cargo pumps are operated at their maximum pumping capacity for an

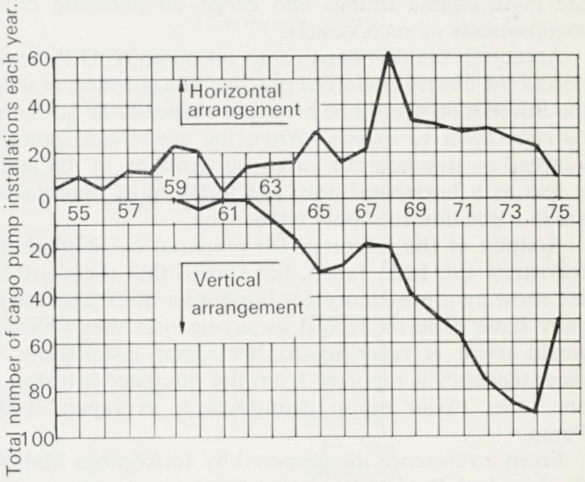


FIG. 1

Total number of cargo oil pump installations per year.

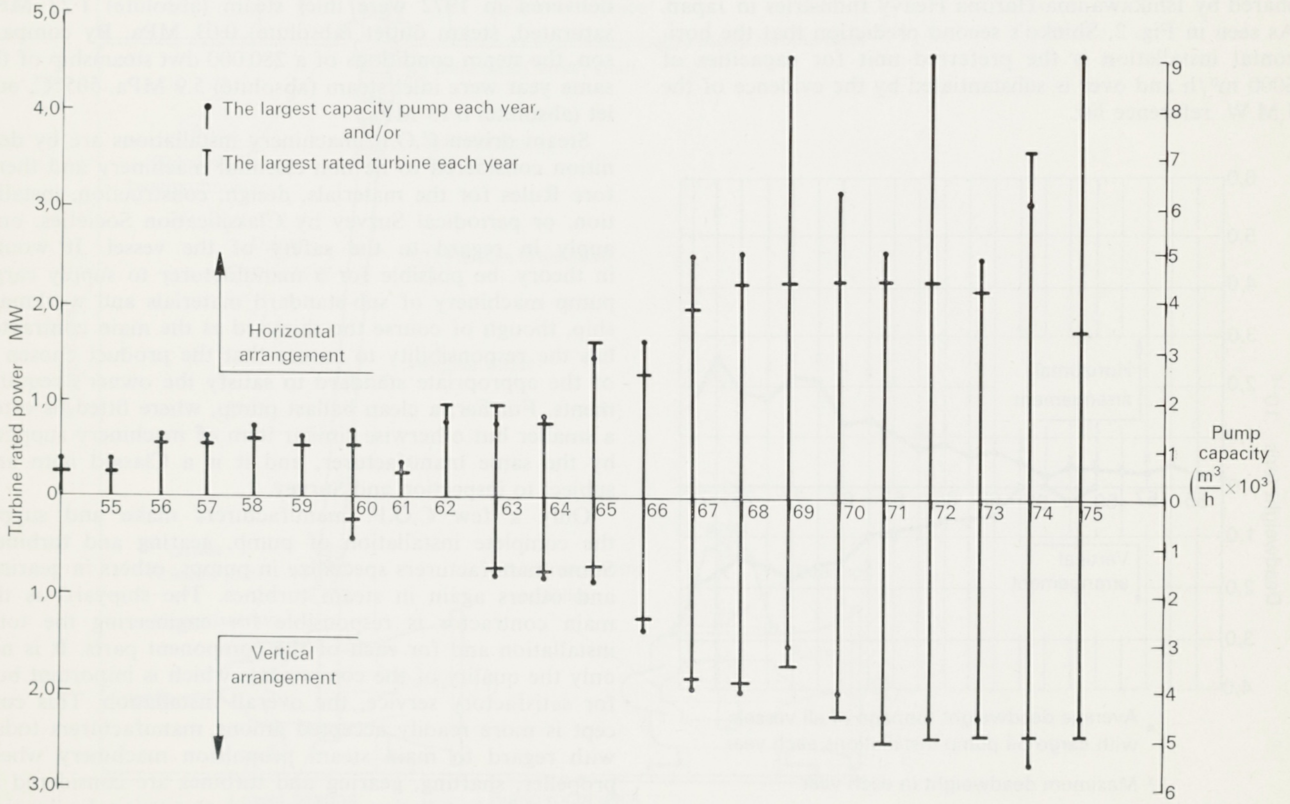


FIG. 2

Largest capacity pump, and/or highest rated turbine per year.

average of about 1 hour per day or 40 hours total in a 40-day round trip, which makes it difficult perhaps to understand why expensive cargo space should be sacrificed to accommodate such machinery when a shore installation could be used on a more continuous basis. There are, however, considerable difficulties associated with any such proposal. Bottom or side discharge pipe connections, for example, would be particularly objectionable with regard to possible pollution hazards.

Although the sizes of crude oil carriers had been steadily increasing up to 1967, closure of the Suez Canal in 1967 provided the necessary impetus to accelerate the growth in tanker sizes, and thus lead to corresponding increases in the main engine ratings and cargo oil pumping capacity requirements of such vessels.

Accepting the on-board cargo oil pump (C.O.P.) installation as the clearly preferred practice, the question of whether the installation should be arranged vertically or horizontally remains open to debate. When the centrifugal pump was adopted as standard for cargo oil handling in the late 50's it was as a horizontal unit, but today most manufacturers are in a position to supply either type.

A study of the reference list prepared by Shinkokinzoka Industries Co. Ltd., Japan, has shown that their customers are showing a preference for horizontal units, and the company have gone on record as saying that they believe the world trend is reverting to horizontal installations. The same tendency is reported from the reference list of at least one other major pump manufacturer in Japan; Teikoku Pump.

From a reference list prepared by Jönköpings Mekaniska Werkstads A.B. (J.M.W.), the inference seems to contradict Shinko's forecast, for, as illustrated in Fig. 1, with the exception of a reversion to horizontal installations in 1967 and 1968, J.M.W's customers appeared progressively to favour the vertical cargo oil pump installation, a view shared by Ishikawajima-Harima Heavy Industries in Japan. As seen in Fig. 2, Shinko's second prediction that the horizontal installation is the preferred unit for capacities of 6000 m^3/h and over is substantiated by the evidence of the J.M.W. reference list.

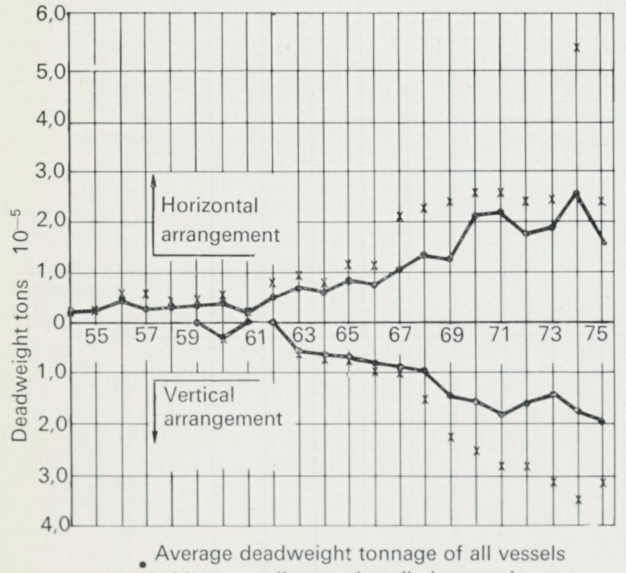


FIG. 3

The average, and maximum deadweight tonnage of tankers per year.

Finally, the inference in Fig. 3, which has also been prepared from the J.M.W. reference list, is that although the largest tankers afloat are equipped with horizontal cargo pumps, from about 1970 onwards vertical cargo pumps were the preferred installation for both V.L.C.C's and products carriers. (Products carriers of some 30 000 dwt reduce the yearly averaged deadweight figures, hence a greater difference between the average and maximum infers more such smaller tankers.)

C.O.P. INSTALLATIONS ON CRUDE OIL CARRIERS

The C.O.P. installation, whether arranged vertically or horizontally, consists essentially of a single centrifugal pump, a transmission shaft which passes through a gas seal in either the deck or a bulkhead, and a driving unit. The drive for both steam turbine and motor tankers is usually through a single reduction gearbox, driven by a steam turbine, though there are instances (Ref. 2) of horizontal cargo pumps being driven through reduction gearing by high speed diesel engines, and the more recent innovation of gas turbine/electric drive for cargo pumps. (Ref. 3)

There are generally four cargo pumps installed on a large tanker, in addition to a ballast pump (which is often a smaller version of the cargo pump), and a stripping pump.

Generally, on a steam turbine propelled tanker, the steam to the C.O.P. turbines may be supplied directly from the main boiler at the full superheat condition, or from a steam/steam generator at lower temperature, and discharging to a separate atmospheric or sub-atmospheric condenser. The motor ship C.O.P. turbine will generally be supplied with steam from an auxiliary boiler at much lower pressure and temperature, discharging to either an atmospheric or sub-atmospheric condenser.

Typical steam condition for a 100 000 dwt motorship delivered in 1972 were inlet steam (absolute) 1·23 MPa, saturated, steam outlet (absolute) 0·03 MPa. By comparison, the steam conditions of a 280 000 dwt steamship of the same year were inlet steam (absolute) 5.9 MPa, 505°C, outlet (absolute) 0·13 MPa.

Steam driven C.O.P. machinery installations are by definition considered to be non-essential machinery and therefore Rules for the materials, design, construction, installation, or periodical Survey by Classification Societies, only apply in regard to the safety of the vessel. It would, in theory, be possible for a manufacturer to supply cargo pump machinery of sub-standard materials and workmanship, though of course the shipyard as the main contractor has the responsibility to ensure that the product chosen is of the appropriate standard to satisfy the owner's requirements. Further, a clean ballast pump, where fitted, is often a smaller but otherwise similar item of machinery supplied by the same manufacturer, and it is a Classed item and subject to inspection and Survey.

Only a few C.O.P. manufacturers make and supply the complete installation of pump, gearing and turbines. Some manufacturers specialize in pumps, others in gearing, and others again in steam turbines. The shipyard as the main contractor is responsible for engineering the total installation and for each of the component parts. It is not only the quality of the components which is important but, for satisfactory service, the overall installation. This concept is more readily accepted among manufacturers today with regard to main steam propulsion machinery where propeller, shafting, gearing and turbines are considered as a complete installation with regard to torsional vibration characteristics, and line shaft alignment and whirling, etc., but it is every bit as important with regard to C.O.P. installations also, especially when it is realized that C.O.P's

are rapidly increasing in capacity and that the steam turbines which will power a single C.O.P. in the late 70's will have nearly the same power as the main propulsion machinery of the earlier oil tankers, namely 8000 s.h.p. (5·9 MW).

SOME DESIGN ASPECTS OF VERTICAL CARGO OIL PUMP INSTALLATIONS

The accompanying diagram (Fig. 4) is a typical layout for the vertical C.O.P. installation, consisting of: —

(1) Pipework to and from the steam turbine casing.

(2) Stop valve and governor valve.

(3) Turbine casing.

(4) Turbine Curtis wheel. (Usually a two-row velocity compounded stage).

(5) Turbine rotor and gear pinion combined, with overspeed trip on bottom end.

(6) Gear casing and lubricating oil sump combined.

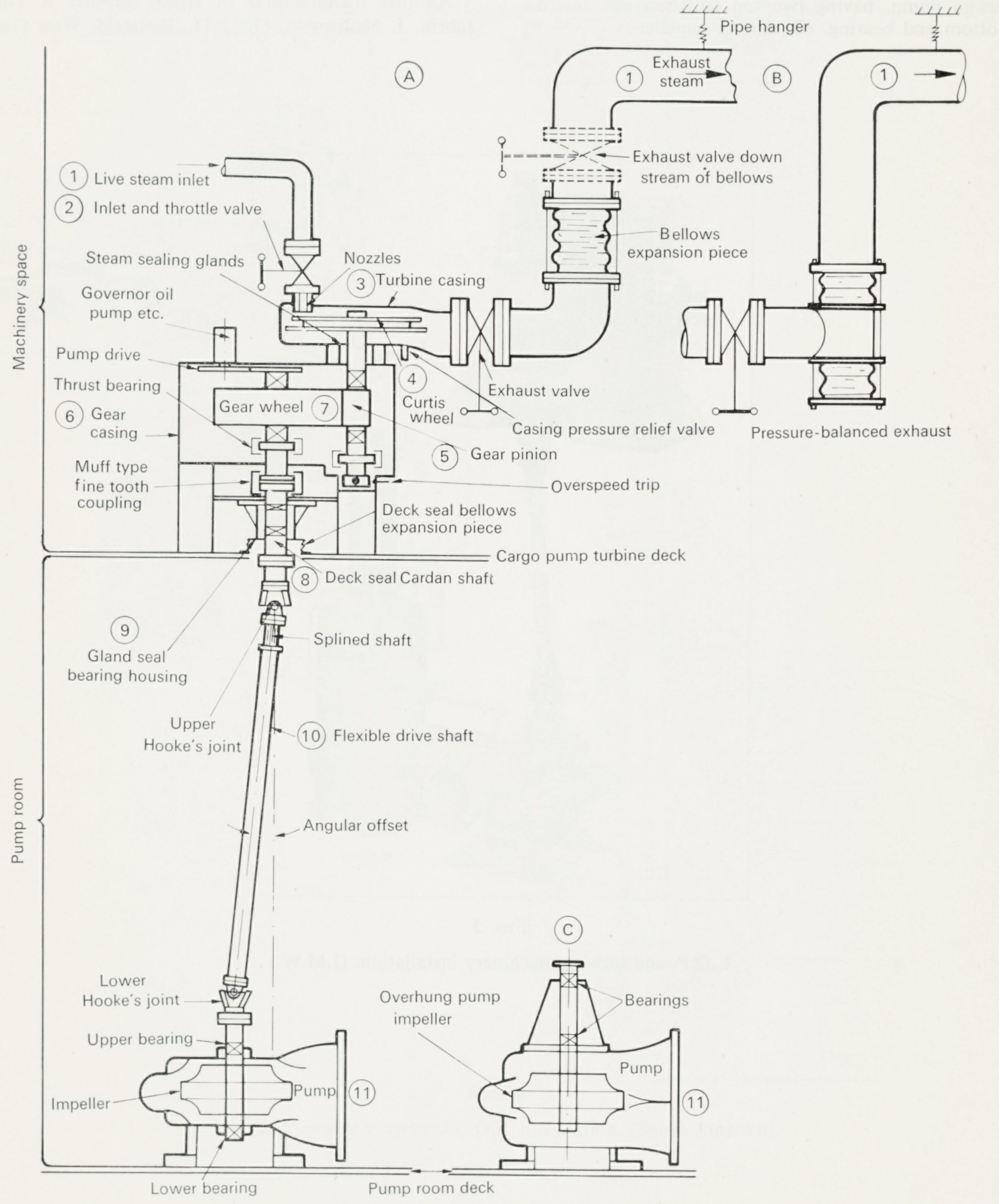


Fig. 4

Typical layout for the vertically arranged C.O.P. installation.

(7) Main gear wheel, with drives for lubricating oil pump governor oil pump, and speed governing mechanism.

(8) Deck seal Cardan shaft, having fine-tooth coupling onto gear wheel drive at top end, and solidly bolted to Hooke's joint coupling of drive shaft at bottom end.

(9) Deck seal and Cardan shaft bearing support.

(10) Drive shaft having two Hooke's joints, and axially splined floating joint at top end.

(11) The vertical cargo oil pump, usually a 'double' or balanced impeller, supported on bearings top and bottom or the recently introduced J.M.W. barrel design pump, having two top end bearings, but no bottom end bearing. (Overhung impeller.)

Examples of the machinery and pump installations supplied by Jönköpings Mekaniska Werkstads AB, Sweden, and Shinkokinzoku Industries & Co. Ltd., Japan, are illustrated in Figs 5 and 6 respectively.

Stal-Laval (Sweden) manufacture only the combined turbine and reduction gearing shown in Fig. 7, which can be arranged horizontally or vertically. Fig. 8 is a cut-away section of the turbine and gears.

Peter Brotherhood Ltd., Peterborough, England, also manufacture the combined turbine and gear installations as shown in Figs. 9 and 10.

Another manufacturer of steam turbines is Turbinenfabrik, J. Nadrowski, G.m.b.H., Bielefeld, West Germany.

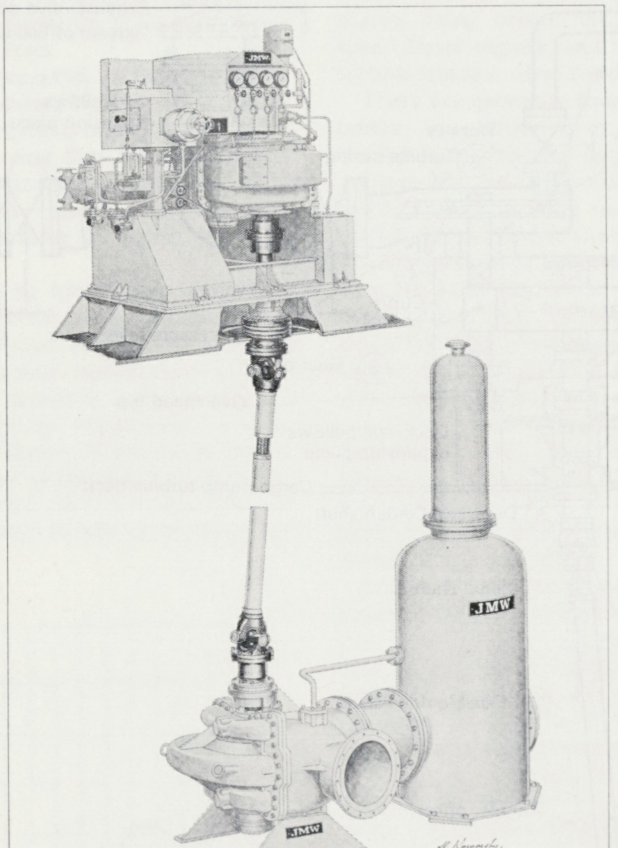
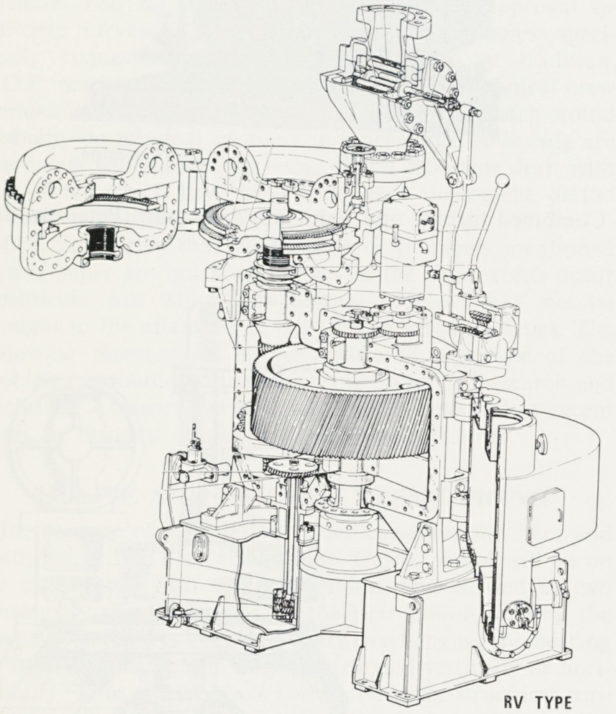


Fig. 5

C.O.P. and turbine machinery installation. (J.M.W.)



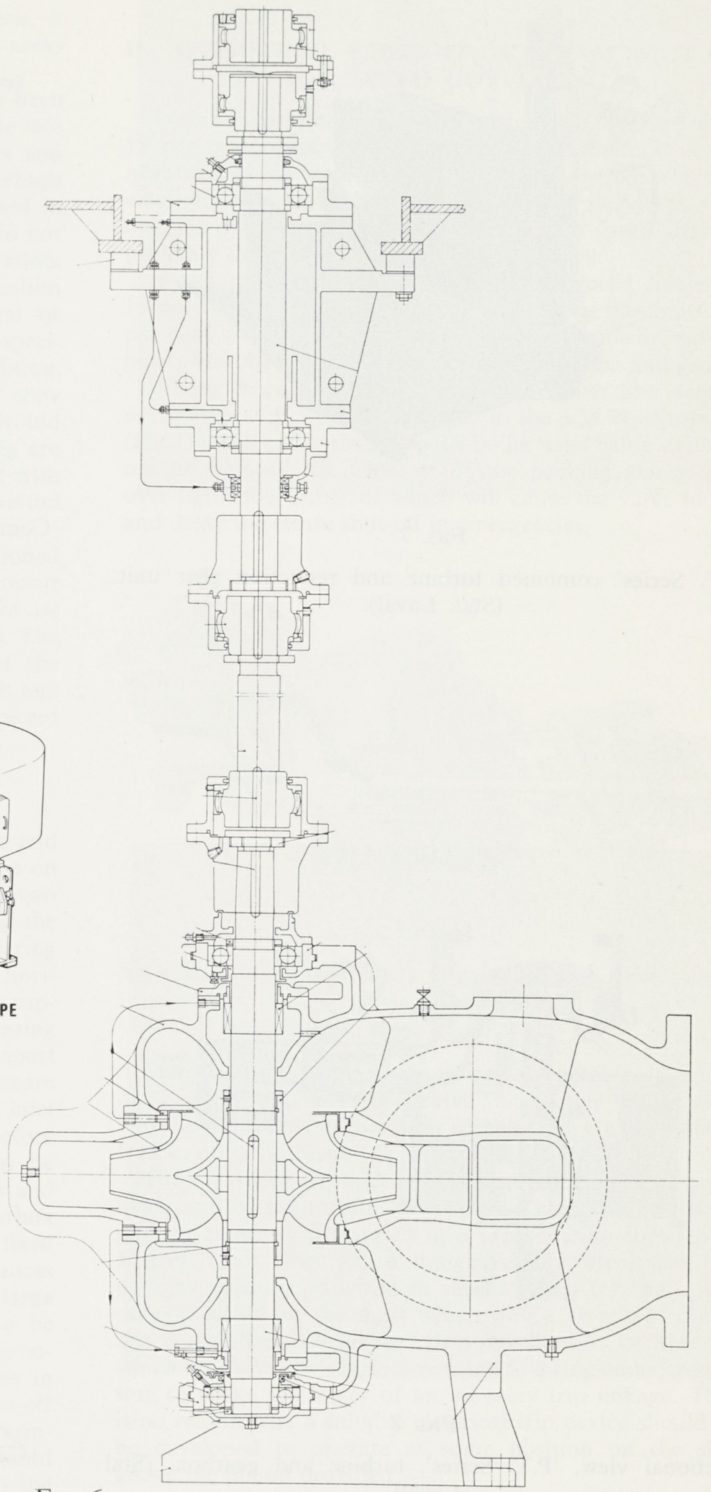


FIG. 6

Sectional view of a vertical C.O.P. installation. (Shinkokinzoku).

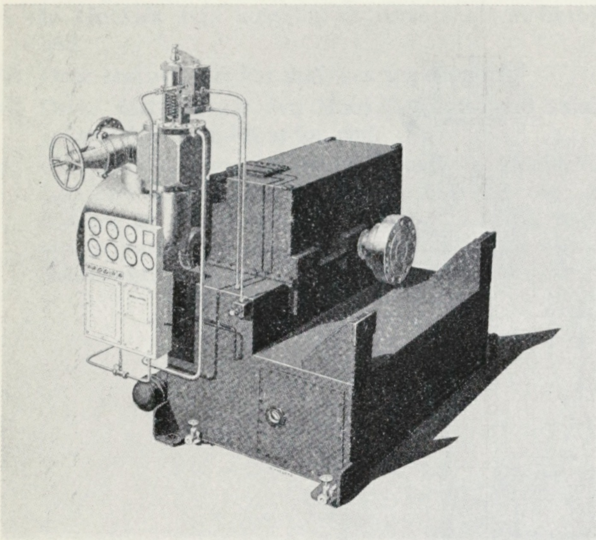


FIG. 7

'P.T. Series' combined turbine and reduction gear unit. (Stal. Laval).

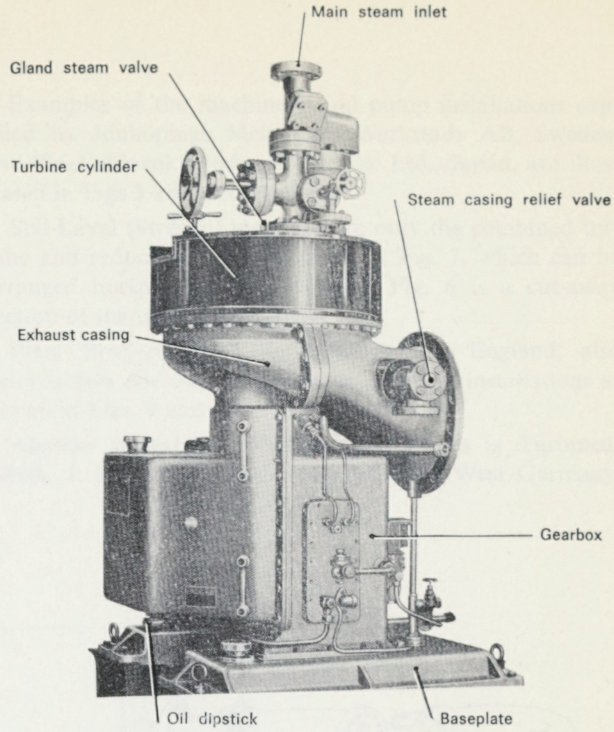


FIG. 9

Combined turbine and reduction gear unit. (Peter Brotherhood).

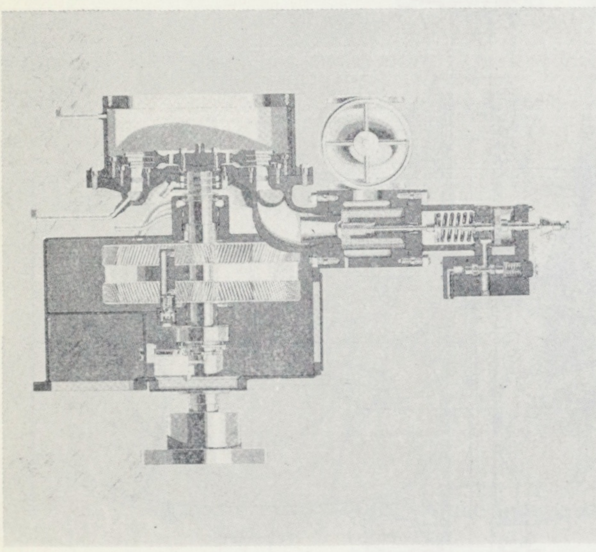


FIG. 8

Sectional view, 'P.T. Series', turbine and gearbox. (Stal Laval).

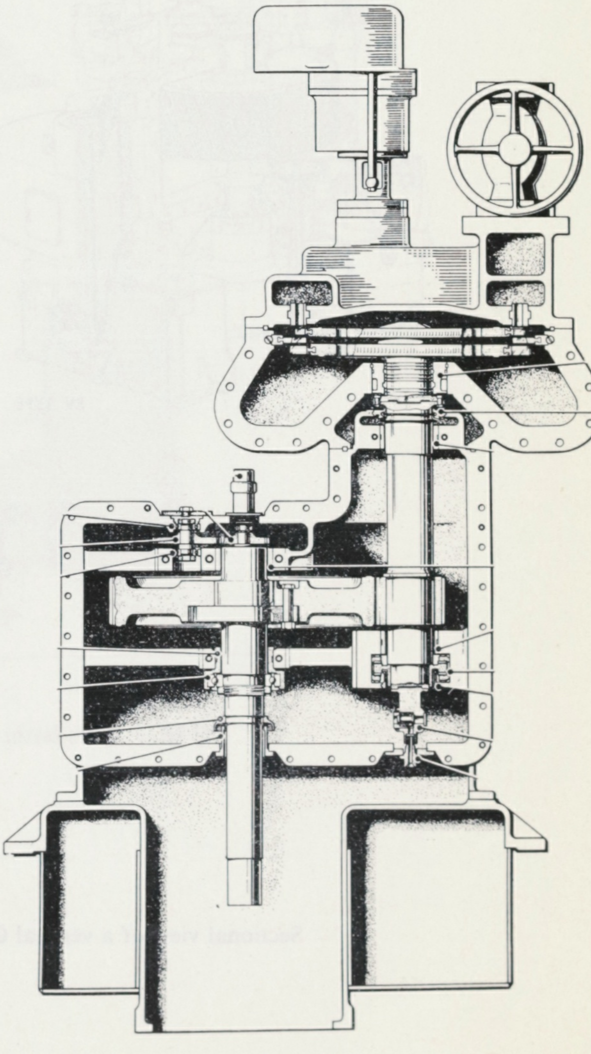


FIG. 10

Sectional view of turbine and gearbox. (Peter Brotherhood).

CARGO OIL PUMP INSTALLATIONS

During the early 70's a number of Shipyards and Shipowners sought the assistance of the Technical Advisory Services Department of Lloyd's Register of Shipping in an advisory capacity, on problems associated with cargo oil pump installations in large crude oil carriers. In the course of investigations on vertical cargo pump installations, it became apparent that there were many more problem areas than had at first been evident.

Undoubtedly a number of the problems may have been the indirect result of the tanker building boom in the late 60's and early 70's, during which time the numbers and sizes of C.O.P's were rapidly increasing, when perhaps sometimes insufficient time and effort was concentrated in applying the principles of good engineering practice to not only the component parts, but also the overall installation. Further, as non-essential machinery, the Classification Societies had no direct involvement in the approval or Periodic Survey of these items of machinery unless specifically requested and paid for. Finally, by tradition, C.O.P. machinery could be operated by non-technical crew members, and the standard of routine maintenance would probably be minimal in many instances. The following are comments which highlight some of the problems dealt with and others which have been considered during the period of active consultancy.

It is very likely that the number of problems mentioned in this paper are but a small part of the total which occur worldwide, but which, for several reasons, may not be brought to the attention of the Classification Societies. The following therefore is in the nature of a review of the problems encountered for which the Society's Research and Technical Advisory Services have been requested in recent years, by Owners, Shipyards, and Manufacturers.

I. LIVE AND EXHAUST STEAM PIPING

In the type of installation under consideration here and ilustrated in Fig. 4(a), the turbine 'Curtis' wheel will be on the same shaft with the gear pinion, and carried in two bearings one on each side of the gear pinion, leaving the Curtis wheel and a length of shaft overhanging the bearing support. Such a turbine may be arranged vertically or horizontally. Thus the gearbox forms the main structural support for all rotating parts, and although the turbine casing is bolted to the top of the gear casing, its purpose apart from steam containment is to house the rotor shaft steam sealing glands and the fixed blades and nozzles. Steam inlet and exhaust pipes are also bolted to the casing, therefore the strength of the casing/gearbox attachment must be adequate to withstand any combination of pipe forces and couples, for any relative displacement between the gearbox and turbine casing could result in contact between fixed and moving blading (*see* Fig. 4(a)). There were instances where pipe forces, due chiefly to pressurization of the large diameter exhaust pipes, caused excessive moments to be applied to the turbine casing and seating bolts. In one instance it was found that bellows-type expansion pieces in the exhaust steam pipes were fitted upstream of the shut-off valve. The shut-off valve would be cracked open for warming through the turbine, but if the turbine stop valve should then be inadvertently opened up, prior to opening up the exhaust shut-off valve, the bellows could be pressurized. In addition to a possible bellows burst, the moment exerted on the turbine casing caused by the pressure differential acting on the projected area of the pipe could disturb the casing/rotor alignment. The casing relief valve should prevent serious pressurization, but the applied moment due to a pressure differential of one atmosphere in the pipe could in some instances be sufficient to deflect the casing, sufficient to reduce running clearances. A more satisfactory exhaust piping arrangement may be seen in Fig. 4(b).

There were also instances where insufficient attention was paid to correctly positioning water drains and dirt traps, and piping was installed sloping downwards away from drains.

II. OVERSPEED, EMERGENCY STEAM SHUT OFF, AND SPEED GOVERNING

Turbine manufacturers usually specify a 10 per cent to 15 per cent overspeed as acceptable, but there have been instances where these limits have been exceeded, where, for example, a group of overload nozzles is provided, which may be detrimental not only for the high speed shaft but probably the slow speed shaft system as well.

If the turbine rotor critical whirling speed lies barely 10 per cent above the operating speed, then the limit of 10 per cent should not be exceeded, if, momentarily, possible heavy vibration is to be avoided in the turbine and gearing.

Where there is a speed governing valve and separate emergency stop valve (E.S.V.), as in the J.M.W. illustration (Fig.11), there is a possibility of boiler salts being deposited on the stem of the ESV, or of the packing glands being over-tightened, either of which will cause the valve to stick and delay the steam shut-off in emergencies.

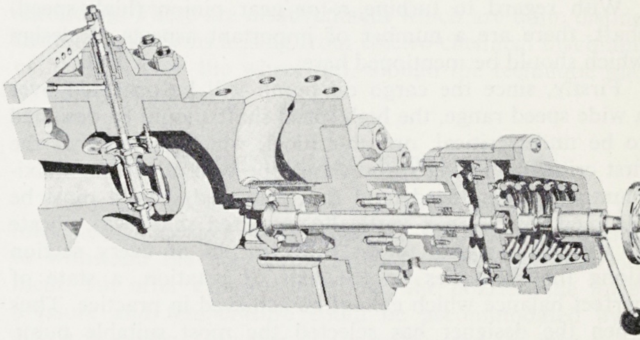


FIG. 11

Section through emergency stop valve and speed control valve. (J.M.W.).

Many manufacturers, it would appear, combine the duties of the ESV and governor valve in a single valve, which may be more reliable for both purposes, but could possibly lead to other problems.

The overspeed trip mechanism is generally located on the lower end of the turbine rotor/pinion shaft (i.e. below the pinion) as neatly illustrated in a cross-section through the J.M.W. rotor (Fig. 12), a design feature which has been perhaps unjustly criticized in recent years, for there is no other position on the shaft which would be acceptable for the conventional overspeed trip mechanism of the type which employs the displacement of a springloaded ring or slug to strike the finger of an oil relay trip linkage. There is no reason why a suitable electrical trip device should not be developed to operate at some position on the shaft above the pinion provided: —

(a) it could reliably withstand the more adverse environment,

(b) it added no mass to the shaft,

(c) it did not reduce the shaft stiffness at any position.

The speed governing mechanism, and the lubrication and relay oil pumps are all usually driven off the top or free end of the gear wheel shaft.

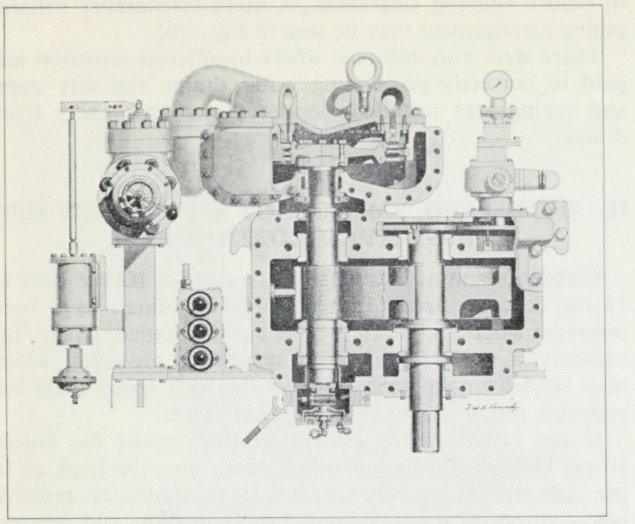


FIG. 12

Section through the turbine and gearbox. (J.M.W.).

III. CRITICAL WHIRLING SPEEDS

1. High Speed Shaft

With regard to turbine rotor/gear pinion (high speed) shaft, there are a number of important aspects of design which should be mentioned here.

Firstly, since the cargo oil pump will be operated over a wide speed range, the high speed shaft should be designed to be under-critical, or sub-critical, which means that the first critical whirling speed should be well above the maximum operating speed (and the overspeed), for it must be assumed that even a vertically mounted shaft will vibrate at its critical speed unless the centroid of every section along the shaft lies on the axis of rotation, a state of perfect balance which cannot be achieved in practice. Thus when the designer has selected the most suitable pump speed and single reduction gear ratio, for the duty required, the turbine/pinion shaft speed will be known, and the best rotor/pinion dimensions can be considered to ensure that the rotor is sub-critical.

Without going into too much detail at this stage it should be understoood that the high speed shaft first critical whirling speed (whether rigidly mounted or allowing for oil film and bearing support flexibility), is dependent upon its dimensions and construction, or in physical terms, the rotor stiffness and mass distribution.

One of these parameters is the overhung mass, i.e. the total mass of the wheel and blades on the top end of the shaft. Other parameters are the length of the overhang, and the diameter of the overhanging shaft, and so on, each of which has a direct influence upon raising or lowering the critical speed.

For example, all other factors being equal, any increase in overhung mass will lower the critical speed. Similarly, increasing the overhang and reducing the overhung shaft diameter will each tend to lower the critical speed. If care is not taken at the design stage in assessing the effect of each of these various parameters, then the critical speed could coincide with the maximum operating speed and severe vibration could result, which can then only be avoided by reducing the operating speed, or by complete re-design of the rotor.

A distinction must also be made between the first critical speed calculated assuming rigid bearing support, and that which occurs in practice due to flexible bearing support, since the latter will always be lower.

2. Low Speed Shafting System

The low speed shafting comprises the reduction gearwheel, a deck seal Cardan shaft, the flexible drive shaft, and the pump impeller shaft. The longest single length of shaft for the vertical installation is the flexible drive shaft which may be from 3 to 4 metres in length having a Hooke's joint flexible coupling at each end as shown in Fig. 5, or the fine-tooth coupling shown in Fig. 6. The gear wheel shaft and pump impeller shafts will probably be offset from each other so that the flexible shaft with Hooke's joint will run with anything from $\frac{1}{2}°$ to 5°, or more, angular displacement at each end. (The barrelled fine-tooth coupling shown in Fig. 6 should not exceed an angular displacement of ± 15 minutes according to Shinko, but they will supply Hooke's joint couplings, Falk couplings, or diaphragm couplings.)

Running speeds for the slow speed shaft can be up to about 1800 r.p.m., being geared down from the turbine high speed shaft in a range of from 4:1 to 8:1, depending on the manufacturer's design.

Calculation of the slow speed shafting critical speeds should include allowances for the flexible couplings, bearing flexibility, and possibly a lumped non-rotating mass to simulate the deck seal bearing housing carrying the Cardan shaft (as seen in Fig. 13).

Such allowances are difficult to estimate with any real certainty since, for example, the deck seal bearing housing flexibility, and lumped mass effect may be dependent upon the engine seating and deck flexibility, etc. Generally, it is advisable to compare calculated results with those measured during tests after installation, and, as for the high speed

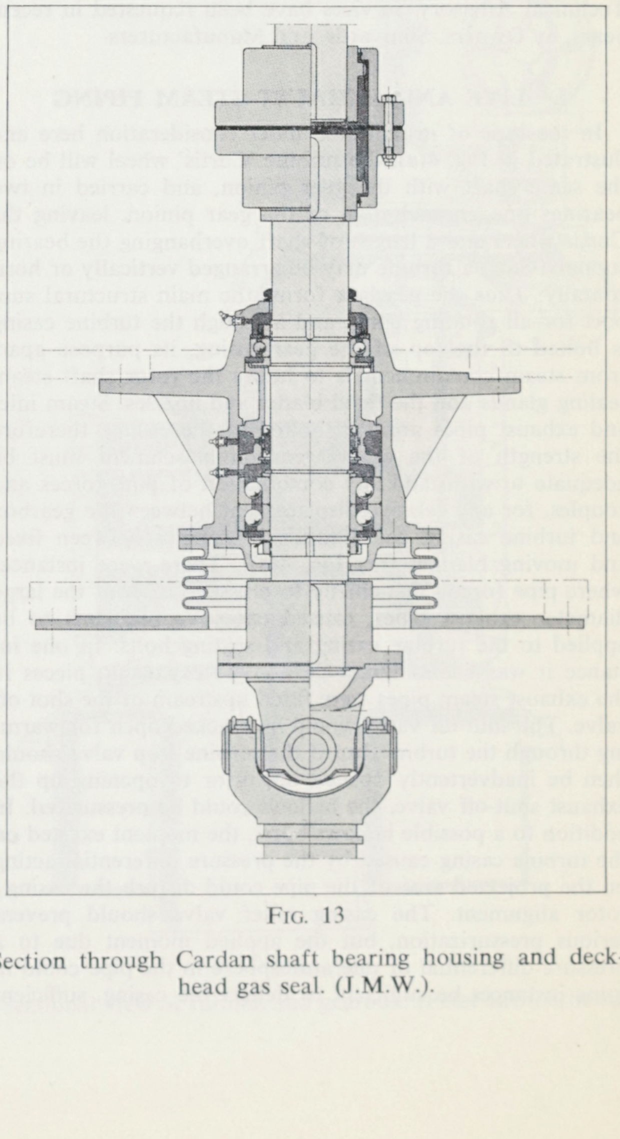


FIG. 13

Section through Cardan shaft bearing housing and deckhead gas seal. (J.M.W.).

shafting, the low speed shaft system should be designed for sub-critical operation.

In addition to the first order (once per revolution), first and second mode critical whirling speeds arising from residual unbalance, there may be a further potential source of excitation associated with the Hooke's joints at each end of the flexible shaft.

Such joints produce a small second order oscillation in angular motion at each joint, presumably accompanied by second order radial force effects of opposite sign, which could excite the shafting system at its lowest natural frequency at a rotational speed below the maximum operating speed. This aspect has not been fully explored to date and it will probably depend upon the support flexibility at each end, length of Cardan shaft, etc., but further examination would require measurements *in situ*.

IV. TORSIONAL VIBRATION

The torsional vibration characteristics of the complete shafting system should be calculated, and the natural torsional frequencies compared with the possible sources of excitation such as gear pitch error, second order Hooke's joint torsional excitation, pump impeller vane impulse, etc.

The torsional excitation due to a Hooke's joint at each end of a flexible shaft should be self-cancelling provided that the angular displacement at each joint is identical, but this may not be the case if, for example, the pump impeller axis and deck seal Cardan shaft axis are not aligned to be parallel when installed with an offset.

Unfortunately, there is insufficient information available at present to assess the magnitude of the various forms of torsional excitation, or the damping characteristics. However, it could be argued that even with the limited data available calculation of the torsional frequencies would be a worthwhile exercise. Case 2 (Section XI) provides an example.

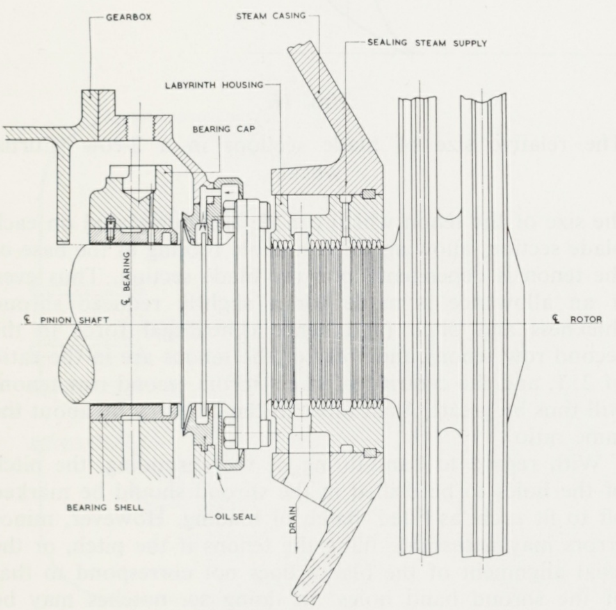


FIG. 14

Arrangement for bolting turbine wheel to gear pinion shaft. (Peter Brotherhood).

V. CURTIS WHEEL ATTACHMENT

All manufacturers of the combined pinion and overhung wheel design of rotor attach the Curtis wheel separately, but there are many variations of the two basic methods of attachment, namely a continuous bolted-on disc (no central hole), or a disc with a central hole keyed to the shaft.

Stal-Laval (Fig. 8) and J.M.W. (Fig. 12) favour bolting the wheel to a spiggoted flange on the end of the shaft within the turbine casing, whereas the Peter Brotherhood design (Fig. 14), takes the flange outside outside the steam space and nearer to the upper bearing. Shinko (Fig. 6) and the Nadrowski design prefer the wheel with central hole keyed to the shaft and held with a locking nut.

The bolted-on wheel in the steam space may result in a marginally greater overhung mass, but in all cases the security of the wheel depends upon the locking nuts. Shinko allow the end of the shaft to protrude, and carry this through a recess in the top half of the casing. It is not a bearing, but could perform as a steady bearing in cases of extreme vibration, or as a fail-safe shaft end retaining bearing.

Nadrowski have reduced the overhung mass of the wheel by electro-chemically machining out the two rows of blades from the solid wheel rim. This has the advantage over the conventional method of fitting separate blades in circumferential grooves in the wheel, of reducing the wheel rim depth and width, which in turn reduces the required thickness of the wheel and hub. There are disadvantages, however, for no shrouding is fitted, which results in windage and tip spillage losses, and from the blade vibration aspect there will be no interface friction damping in the roots or shrouding. These are disadvantages which are quite distinct from the problems arising from electro-chemical machining of blading from the wheel rim, though the technique may be improved in time.

VI. TURBINE CASING MATERIAL

The steam chest containing the steam inlet flange, the inlet steam chamber and nozzles are made from cast steel to withstand the high pressure and temperature environment, but after expansion through the nozzles, the lower pressure and temperature could be accommodated by a cast iron casing. However, if for any reason the inlet valve were fully open, and an exhaust valve shut, then theoretically, if the relief valve did not operate, the casing could become fully pressurized, and may not be able to withstand the full steam pressure.

There is a further possible hazard however, one which should always be carefully considered, and that is the physical containment of moving blades which break off, a wheel which detaches from the shaft, or a burst wheel due to excessive overspeed. If there is any doubt about containment then preferably the casing should be made of cast steel having the required strength and design to absorb completely the energy of the wheel and contain the wheel without fracture, under any of the contingencies mentioned above.

VII. TURBINE BLADES, TENONS AND SHROUDING

There have been a number of instances of blade failures due to metal fatigue. Such failures have been either in the root, or the tenon of the blade. Loss of a blade, or section of shrouding could result in severe rotor unbalance which could consequently damage bearings, fixed blades, gear teeth, etc., if the installation is operating at the maximum speed when the blades fail.

There are probably some instances of blade failure from one cause or another, for which the Society's Technical Advisory Services have not been requested, but of those investigated on behalf of owners or manufacturers, failures were generally confined to the second row of blades of the

two-row velocity compounded stage, and the majority suffered tenon fatigue, though there have also been blade root failures in some instances.

The first row blades for such two-row C.O.P. turbines are generally short and stubby (low aspect ratio), having roots which are often longer than the aerofoil section. This makes natural frequency prediction somewhat hazardous if the point of root fixity at various operating speeds is not known. The sketch in Fig. 15 illustrates the typical proportions such a blade could possess. After the blades are assembled in the wheel, a wedged closing piece tightens the abutment of blades circumferentially. When the wheel is rotating at high speed the centrifugal force of the blades acting on the root lands tends to force open the wheel rim, but should be prevented from doing so by the side lugs on the blade root. In this way an effective 'clamping' should result between wheel rim and blade. The effective 'free length' of blade will then include the aerofoil length and part of the root to the side lugs (A). If at lower rotating speeds the rim is not opened out sufficiently to effectively clamp the root side lugs then the effective blade length is extended down into the root lands, to (B). In the proportions shown in the sketch, such a change of effective length would reduce the lowest natural frequency of vibration by about 50 per cent, and reduce the higher mode frequencies by similar amounts. Natural frequencies are in the Kilo Hertz region. A further complication is introduced if any of the points of 'fixity' of the root are considered as having a finite stiffness rather than being treated as rigid.

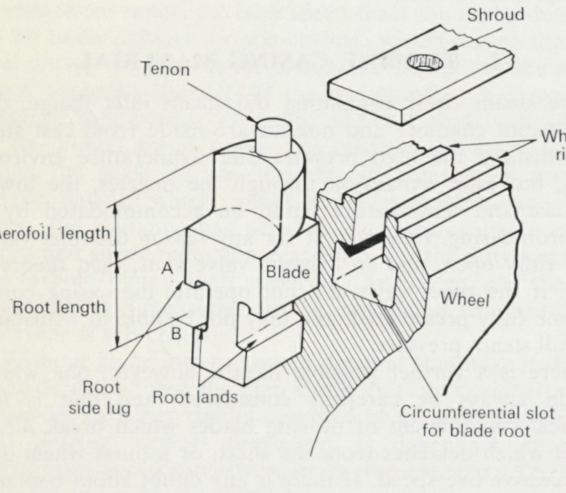


FIG. 15

The first row blade in a 2-row 'Curtis' wheel.

Finally, the interaction between wheel modes and packet modes can sometimes be significant depending upon their relative stiffness and mass distribution, etc.

Steam is admitted to the first row blades through groups of nozzles which occupy an arc of two or possibly three quadrants known as 'partial admission'. The significance of which, in relation to the cyclic loading imposed upon the moving blades, is far more severe than the 'full admission' (complete 360° annulus of nozzles), common in multi-stage turbines, since each blade passing into and out of a small arc of nozzles experiences a cyclic variation from zero load to full steam load to zero load again.

Second row blades are usually longer and thinner in sections but the same remarks apply with regard to calculation of natural frequencies on a somewhat reduced scale. The steam admitted to the second row blades passes through an arc of fixed blades attached to the turbine casing but because steam velocities are lower at inlet to the second row blades, the cyclic loading is also lower.

The greater incidence of second row blade failures may seem to contradict the foregoing, but the more robust first row blades are designed for operation at relatively conservative levels of stress because of the more arduous, and less easily predicted conditions prevailing at outlet from the nozzles. Furthermore, the lowest natural frequencies of the first row blades are, almost invariably, higher than the nozzle passing frequency, whereas the longer second row blades of smaller section have lower natural frequencies, so fixed blade passing frequency could coincide with the first tangential out-of-phase modes of packet vibration (the so-called clamped-pinned modes), at or near the maximum speed. This may well explain failure of second row blades at the root, but tenon fatigue failures can be atttributed in most instances to a combination of excitation of a blade natural frequency with a higher mean stress in the tenon. From the sketch in Fig. 16 which illustrates the relative blade sections which may be employed it will be seen that

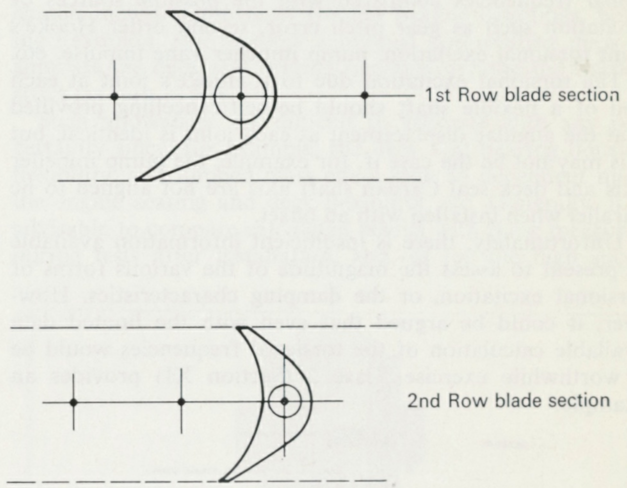


FIG. 16

The relative size of blade sections in a 2-row 'Curtis' wheel.

the size of the tenon which can be accommodated on each blade section, allowing for a suitable footing at the base of the tenon is dependent upon the blade section. Thus even if an allowance is made for a slightly reduced shroud thickness and pitch (and hence centrifugal force in the second row tenon), the areas of the tenons are in the ratio of 2 : 1, and the centrifugal stress in the second row tenons will thus be greater than those in the first row by about the same ratio.

With regard to hand-fitting of the shrouding, the pitch of the holes to be drilled in the shroud should be marked off to fit each 'as-fitted' batch of blading. However, minor errors may necessitate filing the tenons if the pitch, or the axial alignment of the blades does not correspond to that of the shroud band holes. In doing so, notches may be formed in the tenon roots, but additionally, the same amount of metal removed from each of the two sizes of tenons will reduce the effective load carrying area of the smaller tenons by a factor of two compared with the first row tenons. Tenon head formation may also be important, for if the tenon heads are formed by means of a slow hydraulic squeeze technique instead of the time-honoured hammer-blow peening method, the results will be quite different. The rate of plastic deformation is different in each case. The slow squeeze may produce a 'fold' of metal

at the base of the tenon, introducing an effective notch, whereas the hammer-blow method mushrooms only the tenon head. Shroud holes must be correctly chamfered, tenon bases properly radiused, and so on.

VIII. TURBINE ROTOR AND GEAR PINION COMBINED

A number of manufacturers had adopted the shrunk-on gear pinion construction some of which suffered cracking and shaft failures from rotating bending fatigue. These cracks originated from fretting on the shaft surface at entry to the shrink fit, in way of the shaft shoulder fillet. There could be many valid reasons for choosing to build-up the rotor by shrink fitting gear pinions, thrust collars, and the like, and provided that proper attention is paid to the details of design, manufacture, and shrink fitting, such a construction may be satisfactory in some instances.

However, with regard to the turbine/gear pinion shaft critical whirling speed, the shaft stiffness and mass distribution between the bearings are also important parameters. A reduction in stiffness between the bearings will reduce the critical speed. (All other factors remaining constant.) A built-up shaft with shrunk-on gear pinion and thrust collar will in most instances exhibit a lower critical whirling speed compared to a dimensionally identical solid shaft, because the transverse bending stiffness will probably be slightly reduced whereas the distributed mass will be unchanged. Perhaps the most compelling argument against the shrunk-on gear pinion design is the consideration that should the rotor shaft fail in way of the gear pinion the wheel and top end of the rotor could run away, since the overspeed trip is on the bottom end of the gear pinion/rotor shaft, therefore the inlet steam valve would not immediately be shut off.

IX. GEARING

Most manufacturers employ a single reduction spur or single helical gear which makes it necessary to fit thrust bearings on both the high speed and low speed shafts.
The rotor/gear pinion shaft and the main gear wheel shaft bearings may be a conventional shaft collar and thrust pads located below the bottom journal bearing, or in some designs the journal bearing sides act as thrust surfaces.

It will be seen from Fig. 8 that the Stal-Laval C.O.P.T. reduction gears are the double/helical type, consequently, the turbine rotor/gear pinion shaft has no thrust bearings, the axial location of both the main wheel and pinion/rotor rotating elements being maintained by the main wheel thrust bearing.

X. LOW SPEED SHAFTING SYSTEM

Referring to the sketch in Fig. 4 of the vertical C.O.P. installation it will be seen that the low speed shafting system comprises four separate shaft elements, viz: —

(Vertical Arrangement)

1. The gear wheel and shaft
2. The deck seal stub shaft
3. The flexible drive shaft
4. The pump impeller and shaft

Whereas in the case of the horizontal C.O.P. installation (Fig. 17) there are only three basic low speed shafts, viz.: —

(Horizontal Arrangement)

5. The gear wheel shaft
6. Cardan shaft with fine tooth couplings
7. Pump impeller shaft.

The deck seal stub shaft (Fig. 13) is supported in a separate housing and runs in ball bearings. The housing

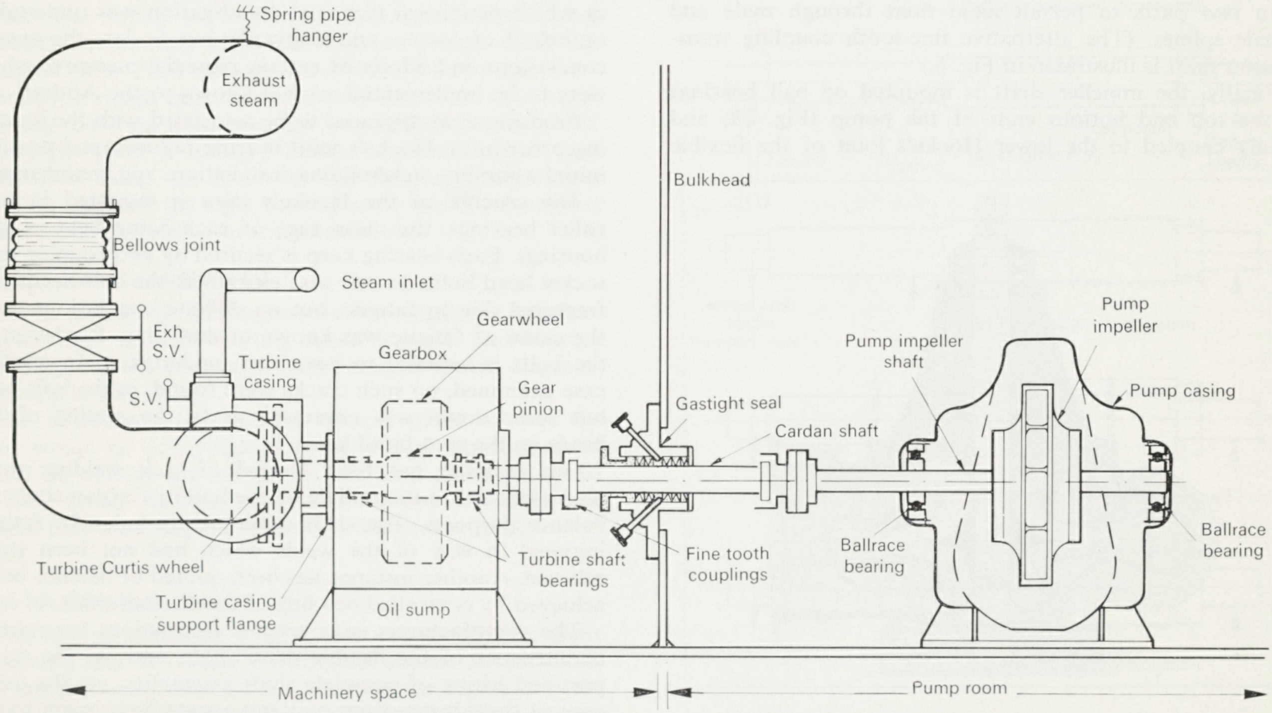


Fig. 17

Typical layout for the horizontally arranged C.O.P. installation.

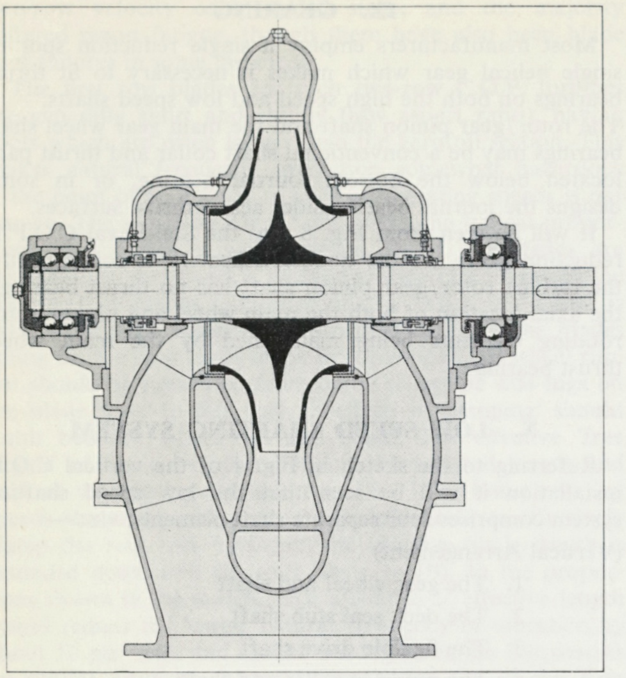


FIG. 18

The double volute cargo oil pump. (J.M.W.).

is flange mounted onto the lower part of the engine seating immediately below the gearbox, and a stainless steel bellows piece is fitted between the ball bearing housing and the deck to provide the gas seal. The stub shaft is flexibly coupled to the gear wheel output shaft through a fine-tooth muff coupling, while the lower end of the stub shaft is solidly coupled to the flexible drive shaft. An alternative type of gas-tight deck seal can be seen in Fig. 6.

The flexible drive shaft is of the 'Hardy-Spicer' type (Fig. 5), having Hooke's joints at each end, and the shaft is in two parts to permit axial float through male and female splines. (The alternative fine-tooth coupling transmission shaft is illustrated in Fig. 6.)

Finally, the impeller shaft is mounted on ball bearings at the top and bottom ends of the pump (Fig. 18) and solidly coupled to the lower Hooke's joint of the flexible

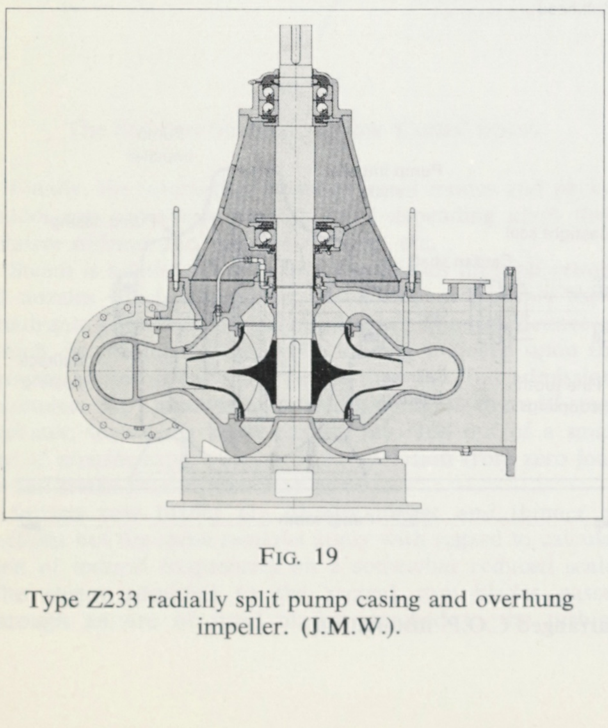


FIG. 19

Type Z233 radially split pump casing and overhung impeller. (J.M.W.).

drive shaft. (Alternatively, the overhung impeller type pump shaft may be supported in two bearings above the impeller, as illustrated in Fig. 19.)

In the horizontal arrangement the Cardan shaft passes through a gas-tight gland seal located in the bulkhead be tween engine room and pump room, one arrangement of which is shown in Fig. 20.

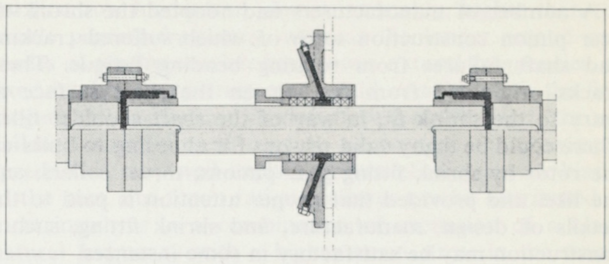


FIG. 20

Bulkhead gas-tight gland seal for the horizontally arranged C.O.P. installation. (J.M.W.).

2. **The deck seal and stub shaft**

The bearing housing (Fig. 13) is effectively cantilevered from the engine seating, so that in addition to ensuring adequate transverse and vertical stiffness for the stub shaft bearing housing and seating as mentioned in Section III, detailed attention should be given to the alignment and lubrication of the fine-tooth muff coupling with sufficient access for alignment and subsequent maintenance. Further, should the gear wheel thrust bearing fail, the stub shaft thrust ring may have to support the gear wheel weight for a period of time, depending upon axial clearances of these components.

3. **The flexible drive shaft**

There are known to have been some failures of this type of shaft manufactured by Gelenkwellenbau Gmbh, Essen, in which peripheral technical investigation was undertaken on behalf of owners and shipyards, but to date the overall conclusions and effects of certain remedial measures which were to be implemented are not known to the Author.

Problem areas appeared to be associated with the balancing procedures, Hooke's joint bearing failures, and possibly initial assembly, newbuilding installation, and maintenance.

The crucifix of the Hooke's joint is mounted in four roller bearings, the outer race of each being held in split housings. Each bearing keep is secured by two counter-sunk socket head bolts. In one case examined, the bolt heads had fractured due to fatigue, but no definite conclusions as to the cause of fatigue was known at that time. Re-design of the bolts is believed to have been undertaken. In another case examined, no such cracks were found, in the bolt head but some doubt was expressed as to the seating of the heads on the spot-faced keep.

One instance has been quoted of tack welding small pieces of steel to the middle of the hollow Cardan shaft for balance purposes. The shaft subsequently failed by fatigue initiated in way of the welds which had not been stress relieved. Another instance has been quoted of balance being achieved by controlled bending of the Cardan shaft.

The manufacturers issue specific instructions for routine maintenance of the flexible drive shaft, and for the transport and lifting of complete shaft assemblies, yet the existence of these instructions may not always be brought to the notice of the owner or the shipyard.

Although it was generally agreed that the axis of the pump and engine drive shaft should be aligned for parallel

offset, the degree of angular displacement thus applied to each joint was open to question. The manufacturers were reported to have stipulated a maximum of 15° angular obliquity but this was considered excessive and values of 3° and 1° were discussed, but no definite conclusions were reached pending measurements *in situ*, particularly to establish the order of magnitude of any changes of alignment due to hull deflections, local distortions, etc., and to determine the influence of the axially splined expansion joint in the shaft.

4. **The pump impeller and shafting**

Two main problems associated with cargo oil pumps have been reported, these being erosion of the impeller due to cavitation, and failure of the pump bearings. It has been suggested that the bottom end bearing of the conventional, vertically arranged pump is not readily accessible for routine lubrication and maintenance and may tend to be neglected, however, it is not known if the incidence of failures supports this supposition.

XI. SOME CASE HISTORIES

The following are brief descriptions of two cases investigated when acting in an advisory capacity for Owners, Shipbuilders, or Manufacturers on various occasions.

Case 1. Products Carrier. 30 000 dwt. Delivered 1972. LR Class

During commissioning tests at the shipyard on the four steam turbine driven cargo oil pumps, severe turbine vibration was reported when the turbines were operating on load between 5600 r.p.m. and 6000 r.p.m. (Service speed 6264 r.p.m., maximum overload speed 7200 r.p.m.)

The turbine manufacturers stated that the first critical whirling speed of the rotor on rigid bearings had been calculated to lie between 7796 r.p.m. and 8300 r.p.m., well above the maximum overload speed of 7200 r.p.m.

Previous vibration had been attributed to loss of turbine blades and, having since strengthened the blades, none had failed. The fitting of anti-whirl bearings had not been successful in limiting vibration, and the pumps could not be tested. The Yard requested the Society's assistance.

Briefly, a rough calculation had indicated that the critical whirling speed on rigid bearings was probably nearer 6500 r.p.m., therefore a detailed computer-aided analysis was undertaken using the Society's shaft whirling programme, LR 272, for both the rigid bearing case, and for a number of different 'composite' bearing flexibilities.

Original Rotor. Analysis (a)

Curve (a) in Fig. 21, represents the calculated relationship between critical whirling speed (r.p.m.), and bearing flexibility, for the original as-built rotor, as shown diagrammatically in Fig. 22(a).

The object was to raise the critical speed so that the rotor would be sub-critical, i.e. the critical whirling speed would lie outside the maximum overload operating speed of 7200 r.p.m., but the number of options available was limited unless a complete turbine re-design was to be undertaken.

First Modification. Analysis (b)

The first modification (Fig. 22(b)) was to increase shaft diameters, and to change the original shrunk-on pinion to one integral with the shaft but retaining the separate thrust collar. Only a limited increase in diameter of the overhung shaft end was possible since the steam sealing gland housing could not be changed. The first modification raised the rigid bearing whirling speed by some 3000 r.p.m., curve (b) in Fig. 21, which appeared to be a satisfactory solution to the problem.

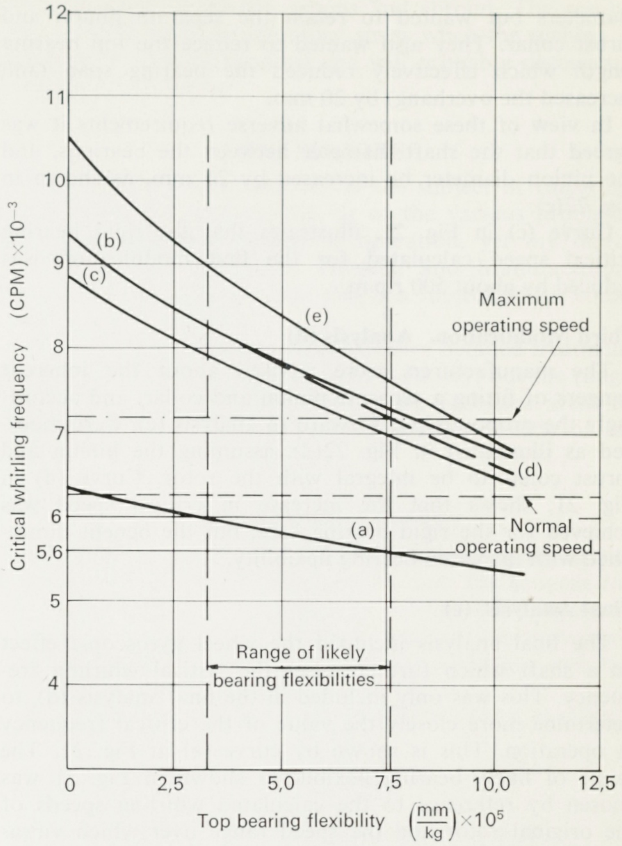


FIG. 21

Relation between the turbine/gear pinion shaft critical whirling speed and bearing flexibility.

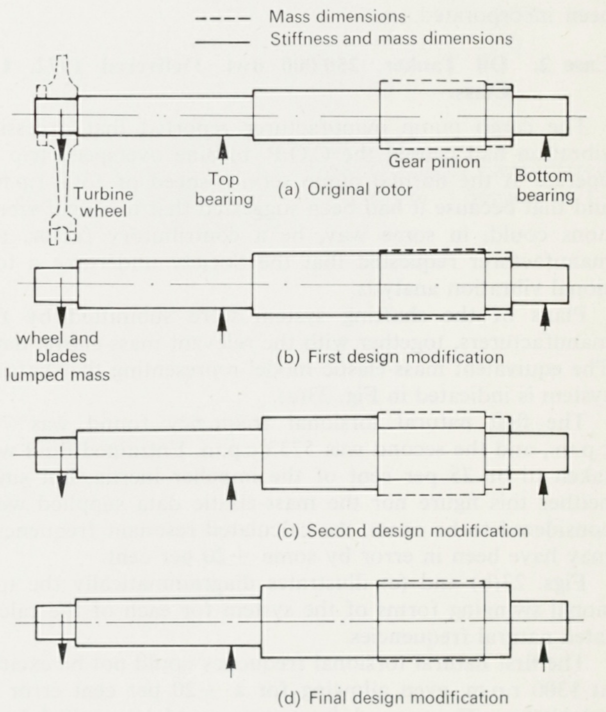


FIG. 22

Representation of the shaft dimension modifications chosen for analysis in Fig. 21.

Second Modification. Analysis (c)

The manufacturers agreed to the proposal to increase diameters but wanted to retain the separate pinion and thrust collar. They also wanted to reduce the top bearing length which effectively reduced the bearing span (and increased the overhang) by 20 mm.

In view of these somewhat adverse requirements it was agreed that the shaft diameter between the bearings, and the pinion diameter be increased by 20 mm, as shown in Fig. 22(c).

Curve (c) in Fig. 21, illustrates that the rigid bearing critical speed calculated for the first modification was reduced by about 500 r.p.m.

Third Modification. Analysis (d)

The manufacturers were advised about the inherent dangers of fitting a separate pinion and collar, and accordingly the proposals put forward in analysis (c), were modified as illustrated in Fig. 22(d), assuming the pinion and thrust collar to be integral with the rotor. Curve (d) in Fig. 21, shows that the increase in critical speed was achieved for the rigid bearing case, but the benefit diminished with increased bearing flexibility.

Final Analysis. (e)

The final analysis included the wheel gyroscopic effect on a shaft which further raises the critical whirling frequency. This was only included in the final analysis (d), to determine more closely the value of the critical frequency in operation. This is shown by curve (e) in Fig. 21. The range of likely bearing flexibilities shown in Fig. 21 was chosen by reference to the calculated whirling speeds of the original rotor and the speed range over which vibrations had been measured.

Vibration measurements carried out by the shipyard indicated a reduction from about 20 mm/sec (R.M.S.), at 6500 r.p.m. for the original rotor to a maximum of about 2·0 mm/sec (R.M.S.), after the design modifications had been incorporated.

Case 2. Oil Tanker. 250 000 dwt. Delivered 1972. LR Class.

The cargo pump manufacturer reported that excessive vibration had caused the C.O.P. turbine overspeed trip to operate at the normal pump service speed of 1300 r.p.m., and that because it had been suggested that torsional vibrations could, in some way, be a contributory factor, the manufacturer requested that the Society undertake a torsional vibration analysis.

Plans of the shafting system were submitted by the manufacturers, together with the relevant mass-elastic data. The equivalent mass-elastic model representing the shafting system is indicated in Fig. 23(a).

The first natural torsional frequency found was 741 c.p.m., and the second was 5733 c.p.m. Entrained fluid was taken to be 25 per cent of the impeller inertia, but since neither this figure nor the mass-elastic data supplied were considered to be exact, the calculated resonant frequencies may have been in error by some ± 20 per cent.

Figs. 23(b) and (c) illustrates diagrammatically the torsional swinging forms of the system for each of the calculated natural frequencies.

The first natural torsional frequency could not be excited at 1300 r.p.m. even allowing for a ± 20 per cent error in calculation. The second frequency could be excited by a fourth order vibration at the maximum speed, but as seen from the swinging form in Fig. 23(c), the maximum amplitude occurs at the upper Hooke's joint of the flexible drive shaft. It was concluded that torsional vibrations would not be likely to cause the turbine overspeed trip to operate, and recommended that the rotor critical whirling speed characteristics be examined.

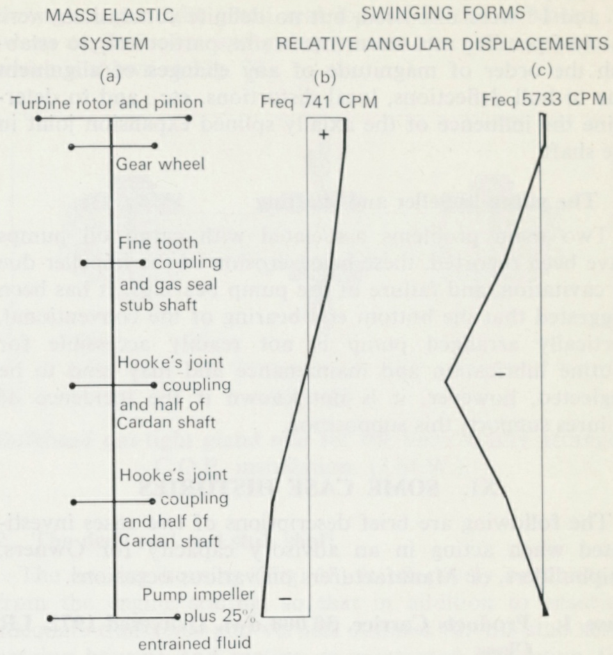


FIG. 23

(a) Representation of the mass elastic system. (b) & (c), Calculated torsional swinging form modes, and angular displacements.

Turbine Blade Vibration Analysis

More recently one cargo pump turbine manufacturer had requested the Society to undertake a computer-aided finite element analysis to determine the blade packet resonant frequencies and the dynamic response characteristics of two prototype blade designs, and one existing blade design.

The finite element technique embodies the 'lumped element' approach, wherein the distributed physical properties of the blade packet structure are represented by a mathematical model consisting of a finite number of idealized substructures (or elements) that are interconnected at a finite number of grid points, to which loads may be applied directly or through offsets, and to which various kinds of constraints may be applied.

The element is a convenient means for specifying many of the properties of the structure including material properties, mass, damping, and stiffness. Each grid point can have six degrees of freedom, viz. three rotational, and three linear motions.

In this instance each of the 29-blade packets was modelled on cylindrical co-ordinates to simulate the curvature of the wheel, and each blade and shroud pitch was idealised by 21 bar elements.

Perhaps the most significant feature of this particular analysis was the development of the facility to include the coupled bending-torsion modes. The blades were tapered from base to tip, but each section was axi-symmetric, and by adaptation of the multi-point constraint method for the axi-symmetric section the use of the relatively simple bar element was possible, the grid points being located along the locus of the twist centre, and the mass centres being offset along the axis of symmetry.

A total of some 75 different resonant frequencies was found by analysis within the range of possible first order exciting frequencies. Plots of three of the normal mode swinging forms for the 29 blade packet are illustrated in the accompanying diagram (Fig. 24) showing plan and side views of the same packet of blades when vibrating in (a) the tangential in-phase mode at 3299 Hz, (b) the tangential out-of-phase mode at 10 020 Hz, and (c) the axial out-of-phase mode at 5951 Hz.

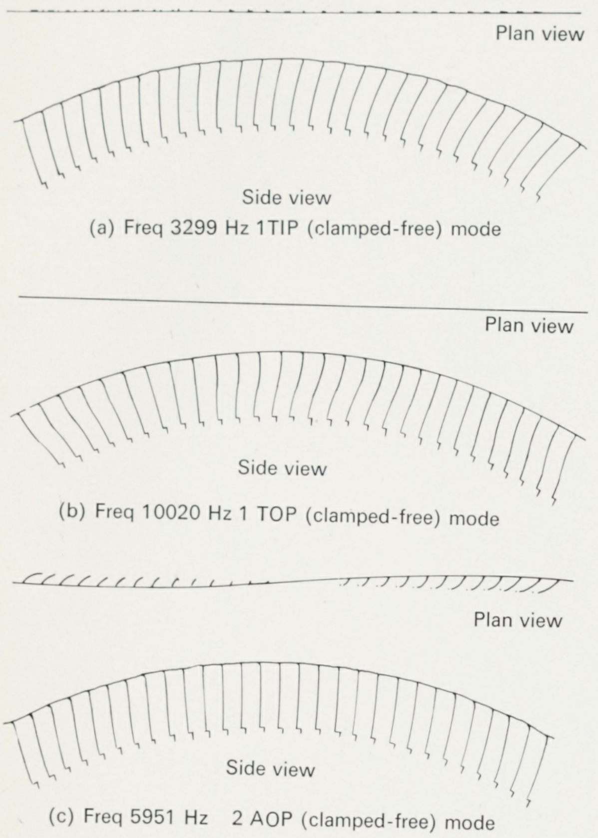


FIG. 24

Plan and side views of the plotted swinging form modes of a packet of 29 turbine blades.

Finally a forced damped dynamic response analysis was undertaken on 64 resonant frequencies to determine the vibratory stress in blades, tenons and shroud. The results indicate high levels of vibratory stress when the turbine is operating at maximum speed and will probably necessitate a design modification.

CONCLUSIONS

Very little has been written on the subject of cargo oil pump machinery installations, or of the various problems associated with their design or operation, yet my direct encounters with Shipyards, Owners, and Manufacturers have led me to conclude that this is a subject which could benefit from discussion.

The time limit does not permit a more detailed treatment of all the component parts of the machinery, but the range of the study is a fair reflection of the problems areas in which the R & TAS Department has been directly or indirectly involved in recent years.

ACKNOWLEDGEMENTS

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SOME FACTORS AFFECTING THE STABILITY OF OFFSHORE SUPPLY VESSELS

D. T. Boltwood

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SOME FACTORS AFFECTING THE STABILITY OF OFFSHORE SUPPLY VESSELS

by D. T. BOLTWOOD

INTRODUCTION

In the last ten years the world has witnessed a tremendous expansion in offshore activities, particularly in the search for oil and gas from beneath the sea bed on the earth's Continental shelves. The rapid consumption of existing energy sources with its economic, political and military implications has provided much of the stimulus for this search.

Discovery has followed discovery and many new areas of activity have emerged, the North Sea, the Eastern Canadian Seaboard, Australia and South East Asia are but a few. This had led to a significant increase in the number and types of offshore drilling installations and associated support craft. Such craft provide an essential service in the maintenance of these installations from shore and without them it is doubtful whether future exploration activity could continue satisfactorily. One such craft contributing to this vital service, often in the most unfavourable circumstances, is the Offshore Supply Vessel, which has built up a high commercial reputation for its toughness, durability and hardworking qualities.

This design of vessel has its origins in the relatively calm waters of the Gulf of Mexico and Lake Maracaibo, Venezuela, where in 1955 the first vessel specifically designed to transport men, material and equipment from shore to the drilling rigs/platforms was put into service. Up to that time such tasks had been carried out by a wide variety of small craft such as shrimp boats, landing barges, air sea rescue boats, etc. The first supply vessel emerged from a cross-fertilization of such craft, its design and role were unique when compared with conventional cargo vessels.

The present day supply vessel still retains the basic configuration of that first design although changes in size, proportions, structural strength, power and manœuvrability have occurred in the last decade.

Such changes were found particularly necessary as the search for hydrocarbons spread into the more hostile environment of the North Sea where gas and oil were discovered in 1965 and 1969 respectively. Deep water exploration and the consequential development of the semi-submersible drilling rig/platform led to a broadening of its role from one of basic supply to include anchor handling

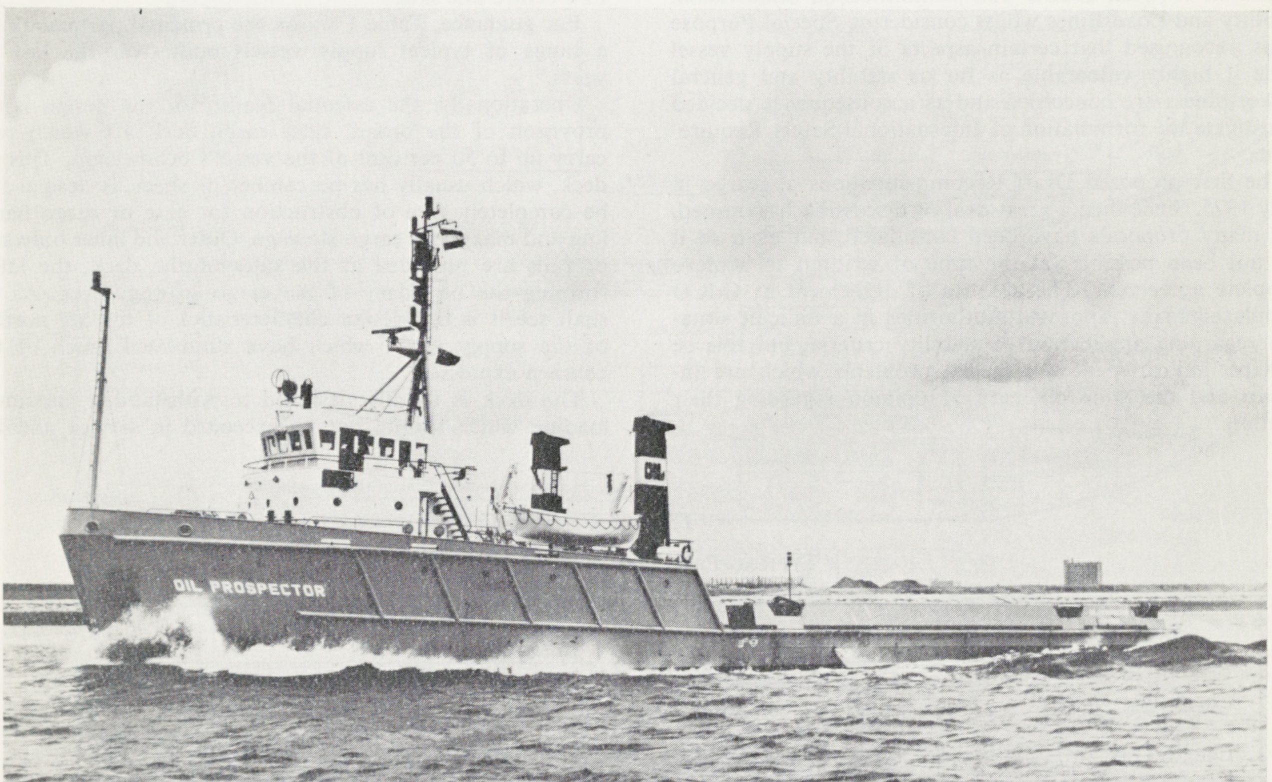


PLATE 1

Typical offshore supply vessel in service. Note the low stern profile.

and towing. Thus emerged what has become known as the tug/supply vessel which is considerably more versatile than the early design and the modern basic supply vessel.

Recently, as exploration has spread into the northern latitudes of Canada and the U.S.A., its role has been further broadened and some designs now include ice breaking capability to deal with the frozen seas encountered in those regions.

Lloyd's Register became involved as early as 1961 when the first supply vessel classed with the Society entered into service in the Gulf area. At that time the fleet size was small because exploration activity had been generally localised, principally to the Mexican Gulf and Venezuela, but by the end of 1973 it had expanded to about 700 vessels. By the end of 1976 it stood at over 1200.

This tremendous growth in the fleet size has not been without incident. Indeed, it has been accompanied by a fairly high frequency of casualty. Many casualties have resulted in the total loss of vessel and more important, loss of life. This situation has led to much concern being expressed by a number of National Administrations and in some cases to the formulation of National requirements covering certain aspects of design and operation. In view of the growth in fleet size and the associated casualty experience the Society, for example, saw the need to develop rules covering the structural aspects specifically for supply vessels. These were issued in February 1973 as Notice No. 2 to the Rules for the Hull Construction of Steel Ships under 90 m in length.

Also in 1973 an IMCO Sub-Committee on Subdivision, Stability and Load Lines whilst considering Special Purpose Ships, recognised that certain aspects of the supply vessel made it highly vulnerable as far as stability and general seaworthiness are concerned and as a consequence decided to instigate the formulation of International Safety Requirements.

The first proposed Draft Recommendations appeared in early 1975. Since then a great deal of discussion has ensued, and many proposals have been considered, but even so it has not been possible (at the time of writing) to achieve complete agreement. The absence of agreement at IMCO has placed many Approval Authorities in a difficult situation regarding application of stability criteria, but this is, perhaps, indicative of the special problems which are involved and the wide diversity of opinion regarding their solution.

The purpose of this paper is to discuss in a general way those factors affecting the stability and general seaworthiness of supply vessels which have caused particular concern and to examine the progress being made regarding the formulation of International Safety Requirements.

Initially, it would be advantageous to describe briefly the supply vessel's general charasteristics, arrangement and function.

GENERAL CHARACTERISTICS AND ARRANGEMENTS

The typical supply vessel is easily distinguished from conventional cargo ships by its unusual profile. Generally, all superstructure and deckhouses are confined to the forward 40 per cent of the vessel whilst the remaining 60 per cent is flat and low in the water. Proportionally, it is small and beamy. Size varies, but typical dimensions are 55 m (B.P.)×12.5 m×5.0 m at a summer displacement of 1800 tonnes. The deadweight to displacement ratio is approximately 0.55 which is small for a cargo vessel.

In comparison with conventional cargo ship forms the length to breadth ratio is small at about 4.4 whilst the breadth to depth ratio is high at approximately 2.5. These abnormal proportions tend to give the hull a shallow box like appearance. Hull forms can be single chine, double chine or round bilge. The lines are characterized by deep V sections forward, a small rise of floor amidships with straight rising buttocks in the after body and the provision of a centre line skeg.

For guidance, Table 1 shows the principal particulars for a range of typical supply vessels built over the last six years.

Operationally, the essential feature of the design is the provision of the broad, clear cargo deck aft which may carry up to 50 per cent of the vessel's deadweight. This aft deck, which usually has no camber or sheer, is designed to be completely free of obstruction for ease of cargo handling and maximum cargo stowage. Outer and inner bulwarks or rails are provided at the sides of the deck, the latter forming the boundary of the cargo stowage area. As we shall see it is the design characteristics of the aft portion of the supply vessel which have stimulated much of the concern expressed.

The deck is usually designed to withstand a maximum loading which should not be exceeded in service and for

TABLE 1

TYPICAL PRINCIPAL PARTICULARS

Year built	1970	1971	1972	1973	1974	1975	1976
Capability	Tug/Supply	Tug/Supply	Tug/Supply	Tug/Supply	Supply	Tug/Supply	Supply
Length O.A. (m)	49.28	53.42	56.39	59.55	35.57	67.00	53.77
Length B.P. (m)	47.21	50.00	55.60	55.15	32.00	58.50	48.00
Breadth Mld. (m)	10.06	11.00	12.50	13.00	7.93	13.70	11.60
Depth Mld. (m)	4.12	4.00	4.80	5.00	3.51	6.74	5.60
Draught Max. (m)	3.55	3.45	4.27	4.36	2.60	5.02	3.86
Displacement Max. Tonnes	1238	1377	1838	2040	339	2720	1654
Deadweight Tonnes	698	757	1053	1091	149	1277	800
Block Coeff.	0.72	0.71	0.60	0.64	0.50	0.66	0.75
BHP	3200	3450	4130	5600	1800	7600	2400
Speed (knots)	12.5	13.0	13.0	13.5	12.0	14.0	11.5

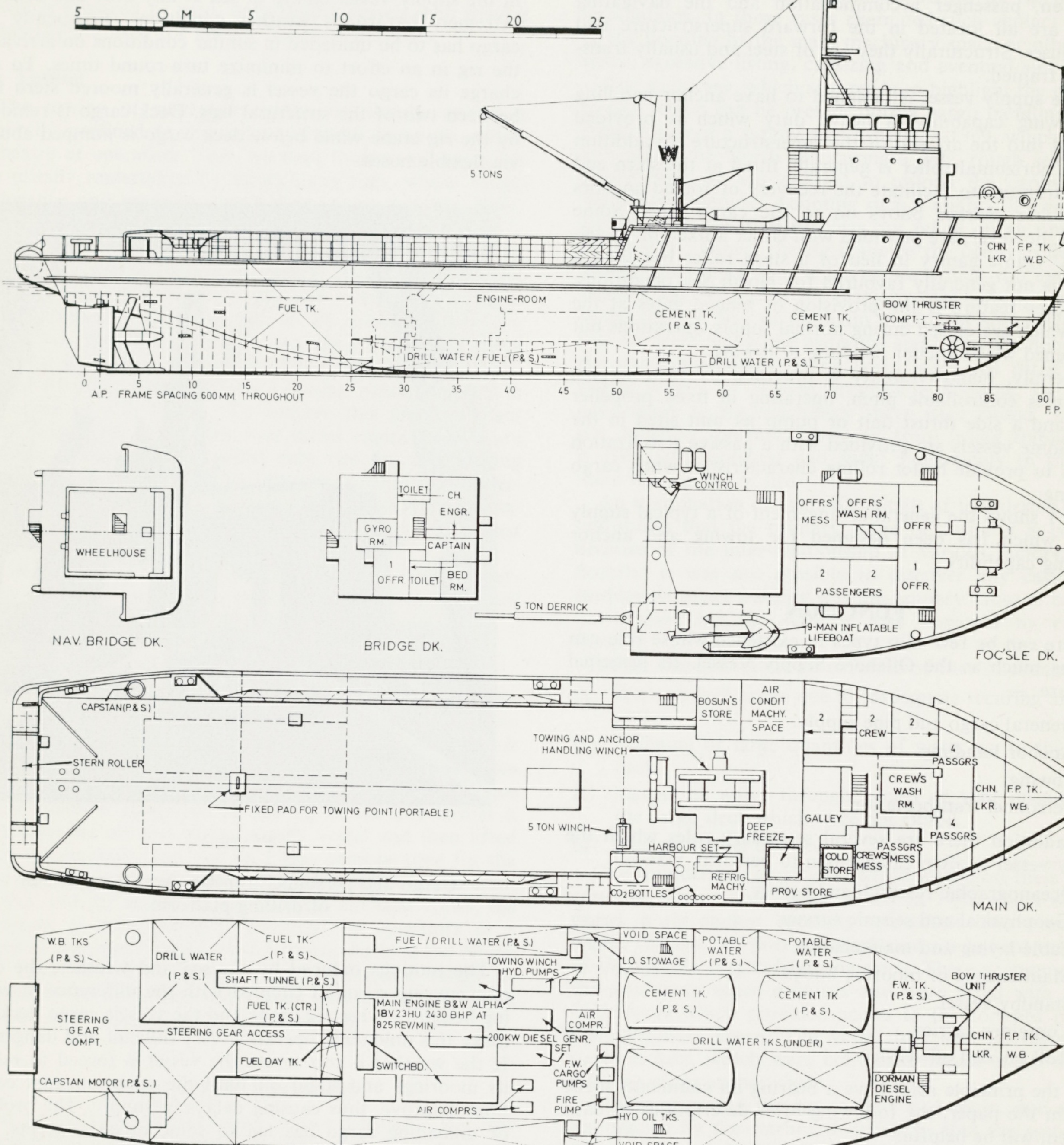


FIG. 1

General arrangement of a typical offshore supply vessel with towing and anchor handling capability.

some vessels may be as high as 5.0 tonnes/m^2. The remainder of the cargo deadweight is carried below the main deck. A separate hold fitted with independent tanks, generally cylindrical, is provided for the carriage of cement in bulk whilst normal integral tanks are provided for various liquid cargoes, e.g. drill water, potable water and cargo oil.

The main machinery space is generally sited at or just abaft amidships, accommodating two diesel engines although more powerful vessels may have four engines. Crew accommodation, passenger accommodation and the navigating bridge are all located in the forward superstructure and deckhouses. Structurally they are of steel and usually transversely framed.

If the supply vessel is designed to have anchor handling and towing capability, a heavy duty winch is provided recessed into the aft end of the superstructure. In addition a large horizontal roller is generally fitted at the stern end of the transom to facilitate the recovery of fouled anchors and anchor marker buoys on to the cargo deck. Some designs, however, are provided with either a fixed or travelling 'A' frame gantry in lieu of a stern roller but such a gantry is not generally favoured for North Sea operations.

Another feature is the provision of hinged gates at the stern which are closed during normal supply operations but are opened whilst anchor handling or towing.

Generally, manœuvrability is provided by twin screws, sometimes controllable pitch, operating in fixed propeller ducts and a side thrust unit or pump jet unit sited in the bow. Some vessels are provided with a passive stabilization system to provide better rolling characteristics when cargo handling.

Fig. 1 shows the general arrangement of a typical supply vessel which has been designed for towing and anchor handling capability.

FUNCTION

There can be few ship types in existence whose role can vary as much as the Offshore Supply Vessel. Its principal roles are: —

(a) General cargo and pipe supply.
(b) Anchor handling.
(c) Towing.
(d) Personnel transportation.

In addition there are several secondary roles which the supply vessel is often called upon to perform, viz: —

(e) Oceanographic research and survey.
(f) Geophysical and seismic survey.
(g) Cable laying and maintenance.
(h) Maintenance of mooring buoys.
(i) Standby vessel.
(j) Container carrier.
(k) Roll on/roll off service.

As the principle roles have a bearing on comments made later in the paper, the following brief descriptions of each of them will be helpful.

(a) General Cargo Supply

In order to keep the drilling rigs and pipe lay barges fully operational, they must be maintained continuously with adequate supplies from shore bases. This involves the supply vessel in the carriage of a large amount and wide assortment of cargoes. For example, to sink a 3 km exploratory well or 'wildcat' in the North Sea, up to 3000 tonnes of consumable material may be required over a 60-day period. This may consist of quantities of cement in dry bulk form, mud products also in dry bulk form, e.g. barytes, drill pipe and marine risers, casing pipe, drilling fluid, drinking water, diesel fuel, refrigerated stores, general provisions and stores and various items of mechanical equipment. Use is made of containers to transport refrigerated stores, general provisions and stores and are carried on the open deck. Often supplies are required urgently by the rig operators who are seeking to prevent lost drilling time which can prove to be extremely expensive. To avoid a loss of contract in a very competitive field this can result in the supply vessel taking to sea at any time and often in extremely hazardous weather conditions. Frequently the cargo has to be unloaded in similar conditions on arrival at the rig in an effort to minimize turn-round times. To discharge its cargo the vessel is generally moored stern first between two of the structural legs. Deck cargo is removed by the rig crane while below deck cargo is pumped aboard via flexible hoses.

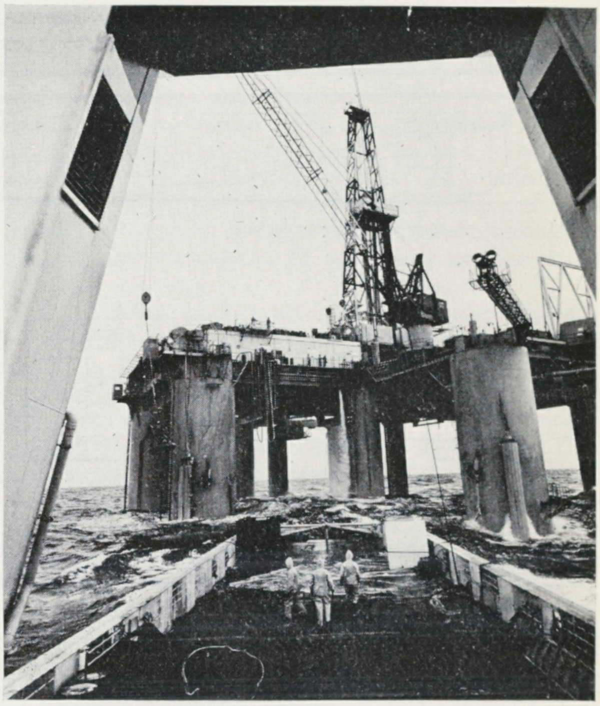


PLATE 2

Cargo and personnel transfer operations in way of drilling platform.

The motions induced by the sea and weather, the close manœuvring required coupled with the difference in working height between rig crane and the vessels cargo deck can make this unloading operation very difficult and dangerous. If the weather is too severe the vessel is forced to release its moorings and stand off until the environmental conditions improve, thus causing delays in supply. The problem of unloading cargo has led to some novel concepts. The moving cargo deck, for example, fitted to some recent vessels is one such concept which attempts to overcome the danger of collision with the platform structure. Normally two vessels will be assigned to one rig to ensure an adequate and continual supply of material.

When supplying pipe lay barges, the deck cargo may be composed only of pipes. Each length of pipe may weigh up to 15 tonnes. Other essential supplies include potable water, fuel oil and fresh water. On average, a fully crewed and operational lay barge may require approximately 40 tonnes of fresh water and 35 tonnes of fuel oil every 24 hours.

(b) **Anchor Handling**

Frequently semi-submersible drilling rigs/platforms, drill ships and lay barges have to move to a new operational location. Prior to movement, the supply vessel's function is to haul each anchor out of the sea bed and return it to the offshore structure. After relocation each anchor is then re-laid in some predetermined position. Each offshore structure may have as many as eight to ten anchors securing it to the sea bed. Anchors generally weigh about 15 tonnes, but may be 20 tonnes on some platforms. Since winch power is required to haul, only supply vessels with towing capability are used for anchor handling purposes.

(c) **Towing**

The towing of semi-submersibles by supply vessels is generally confined to movement between locations within a theatre of operation. Tows between theatres of operation are usually undertaken by ocean-going tugs, whose range, endurance and manning make them more suitable. However, they can be assisted by supply vessels on such tows.

The required towing power tends to vary with area of service. For temperate waters, vessels developing 3000 bhp with a bollard pull of 35 tonnes and towing winch of 80 tonnes capacity would appear to be adequate. For the North Sea, however, latest designs are considerably more powerful and range from 7–9000 bhp achieving a bollard pull of 65–90 tonnes and have a towing winch capacity in the order of 150 tonnes. To achieve this greater power some vessels are fitted with four diesel engines which are all utilized when towing whilst only two are used during the normal supply service. To improve capability to undertake long distance tows bunker capacities have been increased in these latest designs by making fuel and ballast tanks interchangeable.

It is interesting to note that many modern semi-submersibles possess their own propulsion systems but these are not usually adequate for long voyages.

(d) **Personnel Transportation**

This involves the carriage of rig crews, technicians, management, contractors' personnel or supernumeries to and from the offshore installation. Accommodation is usually provided for about 12 passengers. The passengers are often transferred to the rig with the aid of a rope basket which is supported by the rig crane. The basket is landed on the aft end of the supply vessel and then lifted up to the rig deck. This can be a very anxious time for the passengers especially in hostile weather conditions. To combat this and the effects of sea sickness amongst crews, the helicopter is now generally favoured by operators as a means of personnel transportation, particularly in the North Sea. However, if because of fog, inadequate range or economics the helicopter is not used, then the supply vessel must be relied upon to provide this service.

CASUALTY RECORDS

An analysis of the Society's Major Casualty Records indicates that some 32 supply vessels have sunk, the majority of which were total losses, between the years 1968 to 1976. This represents an average of four vessels per year over this period. The last three years have accounted for 15 of these major casualties. Many have involved the loss of life. It is not surprising then that National Authorities have expressed concern over this frequency of casualty and have taken or are about to take action regarding the safety of these vessels. To obtain very detailed information on all these casualties has not been possible for a variety of reasons, but using that available it can be established that they fall into four basic categories, viz: —

Wrecked (vessel to object, reef, platform/rig)	15 cases
Foundered (taking on water, leak, capsizing due to cargo shift)	14 cases
Collision (vessel to vessel)	2 cases
Burnt	1 case

Analysis reveals that two-thirds of the wrecked cases or or about one-third of all major casualties have resulted from contact with the support structure of rig/platform installations. In those cases examined such contact has led to the flooding of the main machinery space or other large space, excessive listing, capsizing and eventual sinking.

One such case, which also tends to highlight the potential hazard which can be present during cargo handling operations, involved a supply vessel, built in 1969, which arrived at a rig in the North Sea during a gale, rough seas and a heavy swell. It was decided to unload the cargo by the 'snatch lift' method. Adopting such a method the vessel did not moor to the platform legs but relied on an opportune moment to connect the rig crane to the cargo to be lifted. Whilst attempting this, a heavy sea struck the vessel and swept it against one of the platform legs causing two holes in the shell in way of the main machinery space. One hole had occurred below the waterline and led to the flooding of the machinery space. Wing tanks were not fitted in the engine room. To prevent progressive flooding, the watertight doors to the machinery space were closed. This did not, however, prevent the vessel from losing stability and it eventually capsized and sank. Fortunately in this case there was no loss of life.

An examination of available details of foundering casualties shows that the primary cause of casualty is complex because of the inter-relationship of various factors. Significantly, it was not possible to discover any case where inadequate intact stability was the primary cause of casualty but rather that the stability was impaired by external factors.

Many cases involved all or some of the following: —

(a) Shift of deck cargo due to inadequate securing arrangements.

(b) Shipping of large quantities of water on to the cargo deck.

(c) Ingress of water through access or vent openings on the cargo deck which were not closed.

(d) Ship personnel failure.

One foundering casualty (1) which highlights these points concerns a tug/supply vessel, built in 1968 which was engaged in rig anchor handling operations. The vessel had engine casings, one port and starboard sited about one-third of the vessel's length from the stern. These accommodated the machinery space exhausts and were fitted with watertight doors which provided access to the forward part of the machinery space. The vessel was also provided with chain lockers, which were converted ballast tanks, for the stowage of rig anchor chain.

Whilst retrieving the rig anchors and chain, one anchor was placed by the starboard casing so that the watertight door could not be closed. In addition the covers to the chain lockers were not secured.

During these operations sea conditions were moderate, but even so water was breaking over the stern and coming through the starboard freeing ports. This water reached the starboard casing and steadily broke over its watertight door coaming, thus gradually flooding the engine room. Heel to starboard, stern trimming and water on deck caused the anchors and marker buoys to move across the deck and the vessel quickly sank by the stern. Seven persons lost thei, lives and two are missing presumed dead.

As perhaps would be expected, heavy weather conditions existed in a large number of the cases examined. The casualties do not appear, however, to be restricted to the more hostile waters, indeed only 22 per cent (seven cases) have occurred in the North Sea during the period considered. About 38 per cent (12 cases) have occurred in the Gulf of Mexico, 10 per cent in The Gulf whilst the remainder (30 per cent) are fairly well distributed amongst other theatres of operation around the world.

One major casualty (2) reported to IMCO in 1974 is important to describe here since it highlights other potential hazards which can exist operationally. It concerns a supply vessel of typical proportions which sank in a severe storm in the Indian Ocean whilst the wind force was Beaufort 9–10. The vessel was steaming at full speed, about 12 knots, in a following sea. This situation had caused significant deck wetness. The deck cargo consisted of 225 tonnes of unplugged pipes. The vessel encountered two quartering waves, the first broke over the stern and flooded the cargo deck. Water entered the unplugged pipes and they shifted partially to one side. Before this water on deck and in the pipes could be completely freed the next wave broke over the stern, struck the wheelhouse and flooded several spaces. The vessel could not recover from the combination of these occurrences and consequently capsized. Five seamen lost their lives.

A detailed analysis of the three casualties discussed above and others raise a number of questions regarding factors which can effect stability and seaworthiness, and these will now be discussed individually.

SUBDIVISION AND DAMAGE STABILITY

The major casualty experience up to the present time would tend to suggest that supply vessels should possess the capability to survive some degree of flooding resulting from shell contact or collision damage. If this fundamental point is accepted then the problem is to decide what this survival capability should be. Some Authorities, however, are not convinced that the introduction of International Regulations covering surviability standards are necessary or indeed timely and call for a detailed analysis of all contact or collision casualties to be made before making any decision.

Certainly, the probability of sustaining contact and collision damage during service would appear to be considerable. An examination of the Society's minor casualty records indicates that supply vessels have a very high incidence of damage due to contact and collision when compared with ships of similar characteristics and operation. Appendix I shows the results of this comparison and suggests that the incidence of damage for supply vessels is over twice that for small dry cargo vessels and approximately five times that for tugs. Some 134 cases of minor damage have been reported during the last two years and significantly, damage due to contact and collision occurred in 73 cases, i.e. 54 per cent of the total. Analysis of the areas of damage indicates that the principal area was the side shell where damage occurred in 70 of these cases. The two most vulnerable areas of the side shell were observed to be the stern and the fore peak which accounted for 42 per cent and 34 per cent of these damage cases respectively. The former perhaps reflects the method of positioning the supply vessel when unloading at the rig whilst the latter suggests the typical damage encountered alongside lay barges. The area of the shell in way of amidships, which would include the machinery and cement hold spaces, sustained damage in 18 per cent of these cases.

It is probable that much of the damage sustained in way of offshore installations and lay barges will generally result from low energy contact and collision since cargo handling operations are carried out when the vessel's speed is zero or very low. The energy induced, therefore, will result principally from environmental conditions and it is conceivable that if the shell is pierced during those operations such energy will not result in extensive penetration of the side shell. This line of thinking suggests that any survivability standards should be based upon considerations of minor or local damage.

On many existing designs even a small, localized penetration can lead to the flooding of a complete machinery or cargo space and eventual sinking since there is no inner structural barrier for protection against ingress of water. The major casualty described earlier is an example of how a small penetration can lead to the sinking of a vessel. Some Authorities have suggested the application of a one compartment standard of subdivision but studies (3, 4) have revealed that in general, existing supply vessels cannot survive the flooding of the machinery space and in some instances the cement hold space and aft cargo tanks. It is doubtful, therefore, that, without radical changes in design, these vessels could meet a one-compartment standard of subdivision. These studies have led to proposals that all new supply vessels should be fitted with wing tanks between the transverse bulkheads of the machinery space.

It is now, however, common practice on new vessels to incorporate wing tanks and double bottoms in way of the machinery space and if the low energy concept is relevant then this will clearly lead to an increased resistance to extensive flooding although it does introduce the problems of unsymmetrical flooding. Fendering along the length of the hull is now also a common feature and this provides added protection against penetration due to side damage. The IMCO Sub-Committee reviewing the question of subdivision and damage stability have generally agreed in principle that supply vessels should be designed to withstand at least minor damage. However, there exists a fundamental difference of opinion on how this should be achieved, and opinions are divided basically into two groups.

One group believes that survival capability could be achieved by a suitable arrangement of longitudinal and transverse watertight bulkheads. A formal damage stability calculation is not considered necessary or indeed desirable. In addition complex difficulties are envisaged in the application of an effective damage staibility standard.

The other group disagrees with this view and sees a need for a rigorous damage stability calculation although appreciating that such a calculation will involve a considerable amount of work.

The main proposal (5) of this latter group presently being evaluated at IMCO specifies stability criteria which must be met after damage to any single compartment (including the machinery space) when the ship is fully loaded with the most unfavourable distribution of cargo. The damage is assumed to occur anywhere between two transverse bulkheads in the longitudinal direction, equal to the depth of the ship vertically and to penetrate 760 mm inboard from the ship's side at the summer load waterline. Main transverse bulkheads are assumed not to be damaged and transverse bulkheads extending 760 mm from the ship's side at the summer load waterline are considered to be main transverse bulkheads. In the final stage of flooding the angle of heel due to unsymmetrical flooding is limited to 15°, but could be 17° if no deck immersion occurs, the maximum residual righting lever should be not less than 0.1 m and the range beyond the position of equilibrium has to be at least 20°.

The final waterline after allowing for sinkage, heel and trim is taken to be below the lower edge of any opening through which progressive flooding could take place. The proposal follows closely the Requirements of the Norwegian Maritime Directorate which were introduced in

1975. The residual stability criteria after damage being the same as those which were adopted by the 1966 International Load Line Convention.

To examine the effect of applying these criteria and damage assumptions to existing designs a damage stability investigation was conducted on two typical supply vessels built during the last five years.

Each vessel was considered to be in the fully loaded condition with maximum deck cargo and have the following particulars: —

	Vessel 1	Vessel 2
Length O.A. (m)	56.51	58.25
Length B.P. (m)	51.33	52.20
Breadth Mld. (m)	12.50	12.80
Depth Mld. (m)	4.80	6.00
Draught. Max. (m)	4.27	5.11
Trim (m)	−0.83	0.12

Damage to each compartment along the length of the vessel was considered individually and the resulting stability analysed during six stages of flooding and comparison with the proposed criteria was made in the final stage of flooding. To assess the importance of wing tanks, two damage cases were assumed in way of the machinery space, i.e. (a) with wing tanks (b) without wing tanks. The results of the investigation are summarized below. Detailed results for the major compartments are shown in Appendix II (a–g).

Vessel No. 1

The proposed criteria could not be met in five cases, two of which would involve sinking of the vessel. Significantly, flooding of the machinery space without wing tanks would cause the vessel to sink but with tanks fitted the criteria could almost be complied with. Sinking also occurred when the aftermost starboard cargo tank was damaged.

Vessel No. 2

The proposed criteria could not bet met in two cases, one of which was the machinery space without wing tanks, however, all other cases were satisfactory.

In the cases of non-compliance for these vessels, it was observed that the value of maximum residual righting lever was the limiting criterion. The angle of heel due to unsymmetrical flooding was generally very small although in one case an angle of 8.5° was realised. The results from this investigation, though clearly not conclusive, tend to confirm those (3) obtained by the Norwegian Authorities, i.e. that supply vessels provided with wing tanks would probably comply with the proposed criteria in the damaged condition. The results also show that division of long cargo tanks aft of the machinery space may be necessary to achieve compliance.

The floodable length curves for the two vessels (Appendix II (h &i)) tend to suggest the difficulty in applying a one-compartment standard of subdivision to present designs.

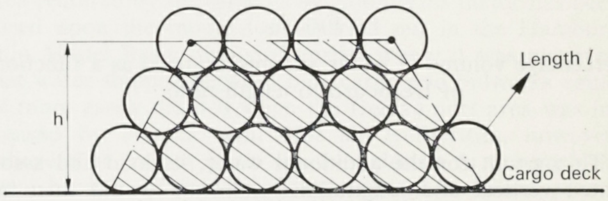
WATER ENTRAPMENT IN PIPE CARGO

The carriage of pipes represents an important part of the vessel's supply function. For example, a typical exploratory drilling well may require up to 500 tonnes of piping over a 40-day period, whilst lay barges will need a continuous supply during oil pipe line laying operations since their pipe storage capacity is small.

Piping used in drilling operations will consist mainly of drill pipe, casings, drill collars and marine risers and will range from about 90 mm to 760 mm O.D. Casings, which provide the essential internal support to the drill hole and a channel for the drilling mud, form the main item of piping. These will be of steel, range from about 180 mm to 760 mm O.D., be supplied in 10 to 14 m lengths and weigh between 0.52 to 5.6 tonnes each for an average 12 m length.

Oil pipe line or line pipe is larger and ranges from about 810 mm to 1120 mm O.D. (steel) and is supplied in 12.2 m lengths. These are much heavier than casings since they are coated with concrete primarily to ensure negative buoyancy when in position on the sea bed and may weigh between 8 and 18 tonnes each. Occasionally 24.4 m lengths are supplied in which case the weight range may be as high as 16 to 36 tonnes each. The carriage of such pipes is not without risk. Pipes are stacked longitudinally between inner bulwarks or rails on the cargo deck with the pipe ends generally open. The small freeboard and aft trim when loaded makes the vessel vulnerable to the shipping of water over the stern, particularly in a following sea. Water can enter and move around such pipes thus becoming momentarily trapped. Following seas of a relatively short period can mean that this water cannot escape before the next wave breaks over the stern and the situation becomes compounded. This additional water on deck clearly leads to an increase in the effective deck cargo load which in turn causes a rise in the vessel's overall centre of gravity and a reduction in statical stability. The casualty mentioned earlier in the paper serves as an example of the possible consequences of water entrapment given a particular set of circumstances. It was estimated after this casualty that as much as 150 tonnes of water could have been trapped in the pipe cargo at the time of capsizing.

The dangers of such entrapment have been acknowledged by a number of National Authorities, indeed some have already included an assessment of its effects on statical stability as part of their requirements. Norway and Australia are two such Authorities. In such requirements loading conditions involving the carriage of pipes include an allowance for water entrapment. IMCO have reviewed this aspect but agreement has been very difficult to achieve. The problem has not been in the principle of taking account of entrapped water but in deciding the volume of water assumed to be trapped. This is not a simple problem since there are obviously many variable factors involved and therefore any assessment can be regarded only as arbitrary. The Australian Regulations assume the volume of entrapped water to be at least one half of the nett area 'A' in and around the pipes multiplied by the length of the pipe cargo, where 'A' is defined in Fig. 2. The c.g. of the entrapped water is taken to be at one-third of the height of pipe cargo as shown. The Norwegian Regulations, however, stipulate a value of 30 per cent of the net volume in and around the pipe cargo whilst the c.g. of the entrapped water is assumed at the c.g. of the pipe cargo (*see* Fig. 3). This latter value has been considered at IMCO but some delegations believe this to be excessive. In addition no account is taken of individual design and operation.



Area for calculation 'A' = Shaded area

Volume assumed entrapped $= \frac{1}{2} \times A \times l$

C.G of entrapped above deck $= \frac{1}{3} \times h$

FIG. 2

Assessment of water trapped in open-ended pipe cargo as adopted by the Australian Authorities.

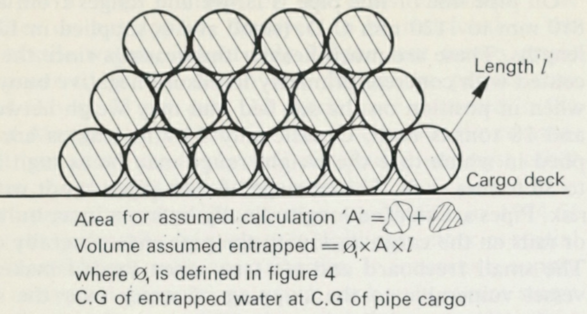


FIG. 3

Assessment of water trapped in open-ended pipe cargo as adopted by the Norwegian Authorities.

A calculation comparing these two requirements was carried out for a deck cargo of 0.34 m O.D. unblanked casings stacked three high between inner bulwarks. The Australian requirements were most severe at 40 tonnes whilst the other was calculated to be 25 tonnes, some 38 per cent less. However, because of the differences in assumed positions of c.g's the transverse vertical moments above the keel were found to be almost identical.

It would seem reasonable that any assessment should consider such factors as variation in freeboard, operating trim, arrangement of bulwarks or rails, height of bulwarks, freeing port area and zone of operation.

A proposal, recently under consideration at IMCO, bases the volume of water assumed trapped upon the assigned freeboard since this factor can be shown to have a significant effect on the probability of shipping water (6). Fig. 4 illustrates the basis of this proposal and as can be seen does not over-penalize those vessels which operate on a freeboard well above the minimum required, although it ignores the other factors mentioned above.

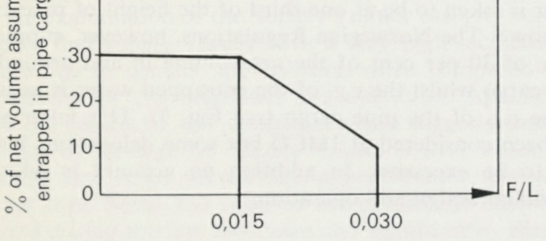


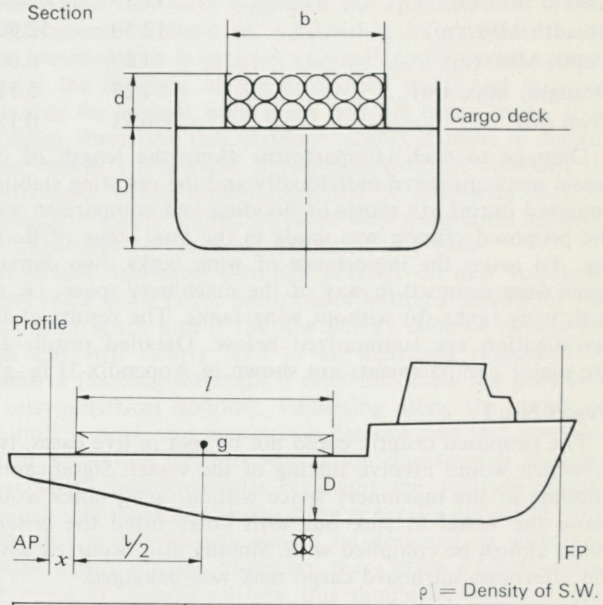
FIG. 4

Percentage volume of water assumed trapped as a function of freeboard to length ratio.

To account for the additional water, its load and associated vertical and longitudinal moments would then be required to be included in the stability calculations when pipe cargo is carried. The liquid free surface of the entrapped water being ignored since it is assumed to be accounted for in the additional load and its transverse vertical moment. This load would not be regarded as part of the deadweight and thus the vessel can exceed its summer displacement in such a condition by the entrapped water load.

This proposal, however, whilst appearing simple does, in the Author's opinion, raise many hidden difficulties. The variation in pipe diameter, wall and coating thickness, length and the assortment of pipe sizes which can be carried at any one time will present the naval architect and in particular the master, with calculative problems when planning loading conditions. Indeed it would seem more appropriate in view of the arbitrary nature of the assessment to consider a simple rectangular idealized envelope of the pipe cargo and take some percentage of this volume. Fig. 5 illustrates this suggestion.

The most obvious way of preventing entry of water into the pipes is by blanking the ends with plugs or caps, but to



Entrapped water	$= \text{Ø}_1 \times l \times b \times d \times \rho$	=
V.C.G. A.B.	$= D + d/2$	=
L.C.G. from A.P.	$= l/2 + X$	=

FIG. 5

Typical assessment of trapped water in pipe cargo.

be effective this must be done positively at both ends of the pipe. The Author has received reports where blanking only at the after end has been carried out. This is considered dangerous since with trim aft the probability of trapping water within the pipes is increased. Pipes have been lost overboard in this way, water flooding the deck, entering the pipes at the forward end, gaining momentum from the vessel's motion as moving aft and gradually forcing pipes overboard. If blanking could be accomplished satisfactorily then the volume assumed trapped would be reduced to that between pipes and it is likely that this will be a matter for each Administration to decide upon.

The Australian Authorities for example take half of the area between pipes shown in Fig. 3 times the length as the entrapped volume acting at $\frac{h}{3}$ when plugs are fitted.

To the Author's knowledge, however, blanking of pipes is not generally carried out and in any case is not thought feasible in practice unless dealt with at the pipe manufacturing plant before transport to the docks. It is understood, however, that some success has been achieved on some Norwegian Flag supply vessels. The plugs used are generally floats of fish nets (plastic balls) which are placed into

the pipe ends at the ship and then inflated by filling with compressed air.

Model tests (7) carried out in 1974 at the Hamburg Ship Model Basin under the sponsorship of the Norwegian Maritime Directorate clearly demonstrated the positive influence that the plugging of pipes can have on the safety against capsizing in certain sea states. The model, a typical supply vessel, was subjected to immediately following or quartering waves and two simulated sea states encountered in the northern part of the North Sea. The results are interesting because the effects of a wide range of factors were investigated, i.e. size of pipe cargo, stern gate, area of freeing ports, poop, length of forecastle. With regard to the size of pipe cargo, the model was observed to be considerably less endangered when loaded with small diameter pipes than with large pipes, when shipping large quantities of water over the stern. In the case of small pipes, the maximum heeling angles were remarkably smaller, sometimes half that recorded for large diameter pipes.

It is interesting to note that certain smaller diameter pipes are buoyant and thus the question of pipe securing arrangements may have to be considered. Casings 178 mm O.D., for example, will possess 33 per cent more buoyancy than weight when blanked.

WATER ON DECK

Clearly, for reasons mentioned earlier the water shipped aboard the cargo deck must be freed from it as quickly as possible. For fully loaded decks this constitutes a serious problem since the water may have to make its way around many obstacles before escaping and if precluded from doing so could lead to water entrapment which is in addition to that already discussed for the carriage of pipes. Empty or partially loaded decks can be even more serious since a greater accumulation and hence weight of water is possible on the deck at any one time.

To assess accurately the amount of water, which may be trapped on deck, for incorporation in stability calculations is even more difficult than the case when carrying open-ended pipe cargo. Indeed, there are so many indeterminate variables involved that the task would appear to be impossible. The vessel's motion characteristics, trim, freeboard, arrangement and height of bulwarks, cargo type, disposition and permeability, freeing port area and environmental conditions will all have a significant effect on the shipping and drainage of water. Among early proposals at IMCO was that which suggested that for empty cargo decks the water should be taken to the top of bulwarks when calculating loading conditions. On some designs this could represent over 300 tonnes and the effect on the resulting statical stability diagram is considerable.

In the case of fishing vessels, the problem of water on deck has led to the introduction of a quasi-static criterion (8) which is used to determine the ability of the vessel to withstand the heeling effects due to water loads in the worst operating loading condition. The heeling lever curve induced by the water load is superimposed on to the vessel's righting lever curve and the residual areas under both curves between specified limits are compared.

The ratio of the latter to the former should not be less than unity. The loading condition assumes that the quantity of water on deck is such that the deck well is filled to the top of the bulwark at its lowest point with the vessel heeled up to the angle at which this point is immersed. In this approach the effects of freeing ports are ignored and both displacement and trim are assumed constant. Whilst clearly a simplified solution, it is thought that such a criterion could be applied to supply vessels where bulwarks are fitted along the cargo deck and where service conditions warrant such considerations.

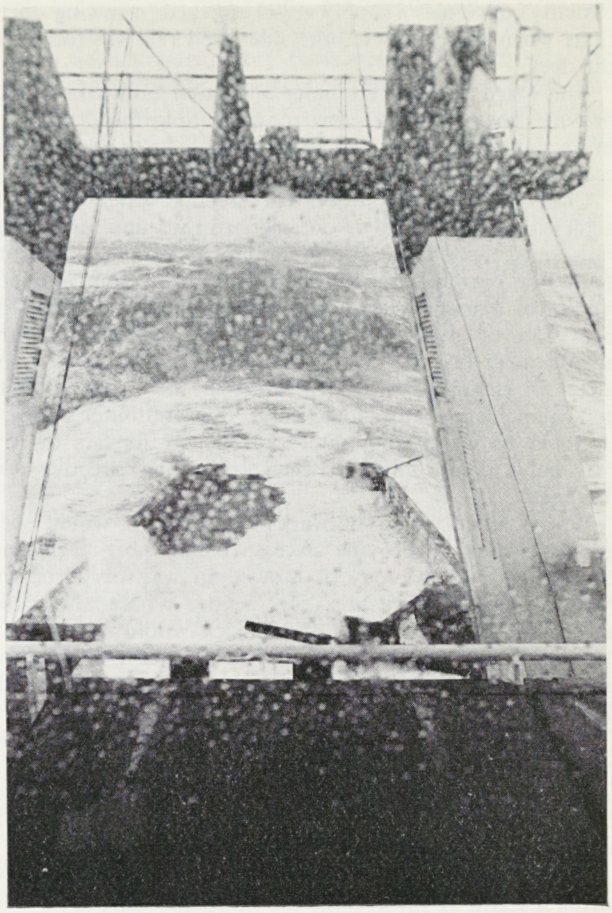


PLATE 3

Water on the cargo deck as seen from the safety of the bridge.

At IMCO, the emphasis has been placed generally on the provision of adequate freeing port area and its careful disposition around the deck, rather than a quasi-static solution. It is likely that such area will have to comply with at least that required by Regulation 24 of the 1966 Load Line Convention, although some Authorities consider this area to be insufficient. The third major casualty discussed earlier is important in this respect, for analysis showed that whilst the available freeing port area was in excess of the Convention, approximately 70 per cent of this area was blocked or closed by deck cargo.

The Norwegian Authorities require 1.5 times the freeing area required by the same Convention. This factor has been based upon the model tests carried out in the Hamburg Ship Model Basin discussed earlier where it was observed that water shipped between pipe cargo and bulwarks could be more easily drained when the freeing port area was increased by 50 per cent. Not all Authorities, however, believe that freeing ports are the solution to the problem since because of low freeboard freeing ports can permit more water to enter than is freed and it is likely national practices may vary on this point.

As indicated the cargo deck is generally fitted with inner and outer bulwarks or rails. The function of the inner bulwarks or cargo rails are to provide a containment boundary for the cargo and they must be powerfully constructed and well secured to the deck to withstand the impact loading which can occur during cargo handling operations. The incidence of damage to bulwarks and rails is very high. One

casualty concerned a supply vessel engaged in towing operations when the complete port side hand rail was torn from the deck.

The space between the inner and outer bulwarks or rails provides an access for crew moving along the deck and a protected area for upper deck fittings such as air pipes, escape hatches, valves, etc.

Clearly the best solution for freeing water would be achieved by the fitting of outer guard rails and inner cargo rails since there exists the minimum of structural obstruction. For reasons of maximum crew protection and possibility of damage to cargo rails some operators favour the provision of inner and outer plated bulwarks but this arrangement has the disadvantage of creating more resistance to escaping water. In practice the most common arrangements are either inner cargo rails and outer bulwarks or inner bulwarks and outer rails which appears to be a reasonable compromise between safety and effective deck drainage.

Where inner plated bulwarks are fitted it would seem good practice to provide excess freeing port area since operationally cargo may be stowed in way of ports thus reducing the effective area for escape. Where both inner and outer plated bulwarks are fitted it would seem advantageous to arrange freeing ports in common longitudinal positions thus permitting continuous drainage across the deck.

The fitting of shutters or flaps to freeing ports is not recommended particularly in service areas where icing is likely to occur because of the probability of malfunction. This can also occur in non-icing service areas unless regular inspection and maintenance is carried out by the crew.

One special area of drainage is the recess or winch space at the aft end of the forecastle. Water can often reach this position in large quantities and scuppers, trunks or similar arrangements will be necessary to free it. In some cases a bulkhead has been fitted across this space at the aft end of the superstructure extension, but to be effective it must be made weathertight. This is difficult because slots have to be cut in the bulkhead and are left uncovered during towing and anchor handling operations to permit passage of wires and chains. In such cases, then, drainage of the winch space would still be desirable.

SOME PRECAUTIONS AGAINST CAPSIZING

The experience gained from casualties has highlighted a number of general precautions which could be taken to lessen the risk of flooding and eventual capsizing resulting from the ingress of water through upper deck fittings. The main precautions are briefly discussed below: —

(a) Accesses on the Weather Deck

Of particular relevance is the access to the machinery space and spaces below the cargo deck. The Society strongly recommends that doors to such accesses should be located in position 2 or if in position 1 should lead to a space or passageway with an internal door of substantial construction with a sill of 380 mm, this passageway being provided with adequate drainage. Generally speaking, in most modern supply vessels the access to the machinery space and spaces below is made from within the forecastle, whilst access to the forecastle is made via a watertight door sited at the fore end of the winch space on the cargo deck. Current thinking is that such doors or hatches which give direct access to the cargo deck should be kept closed during navigation, except when essential for the working of the ship and should always be ready for immediate closure. In addition such doors or hatches should be clearly marked as having to be kept closed except for access.

(b) Ventilation Openings

The principle ventilators on the cargo deck will be the uptakes from the machinery space. To ensure safety the Society requires the coamings of such ventilators to be at least 4.5 m in height when fitted in position 1, and 2.3 m in position 2. They are to be provided with efficient permanently attached weathertight steel closing appliances.

Many modern designs now incorporate these uptakes at the after end of the forecastle with openings at bridge or boat deck level, thus avoiding the damage from cargo handling and water ingress which occurred on earlier designs when uptakes were sited further aft along the cargo deck. Where ventilators are sited on the cargo deck they should be well protected.

(c) Air Pipes

These should, like ventilators be well protected from damage and generally are sited between inner and outer bulwarks or rails and against either. Even in these areas they can sustain damage but the probability is considerably reduced. The Society requires that air pipes on the exposed cargo decks and forecastle decks should be fitted with automatic closing devices and be of a height required by the Load Line Rules although a lesser height could be accepted depending upon bulwark height and protection given to the air pipes. The efficient working of these closing devices is important since air pipes can often be under water during heavy weather conditions. Many cases have been reported where water has entered fuel and ballast tanks via air pipes even though automatic devices had been fitted.

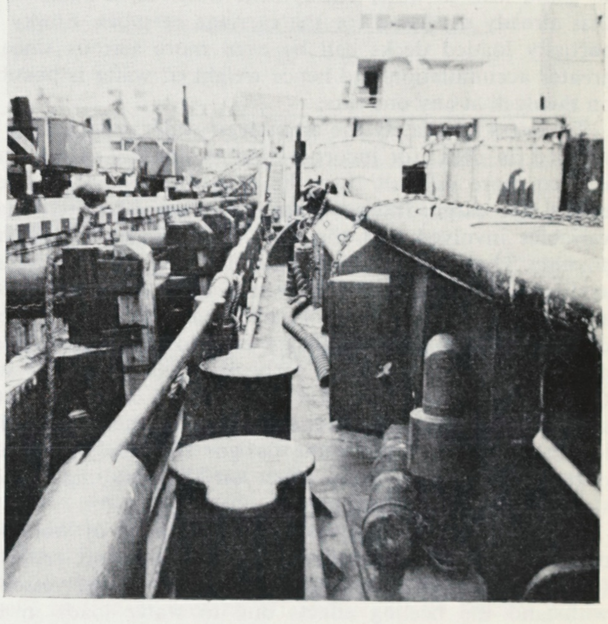


PLATE 4

Access between inner and outer rails. Note the air pipe positioned against inner bulwark.

(d) Cargo Securing Arrangements

Casualty experience has shown that the shifting of deck cargo has occurred in many cases prior to capsizing. Therefore it is important that when carrying deck cargo care is taken by the ship's staff to ensure it is safely secured or stowed to prevent movement before the voyage is commenced.

In practice, securing arrangements are rarely the same, because of the variable nature of the cargo. It will often be lashed to the inner bulwarks or rails by chains or steel wire

ropes which have been led through deck anchoring points and around the cargo. These chains and wire ropes must be strong enough to withstand the environmental forces. Plate 5 illustrates this point.

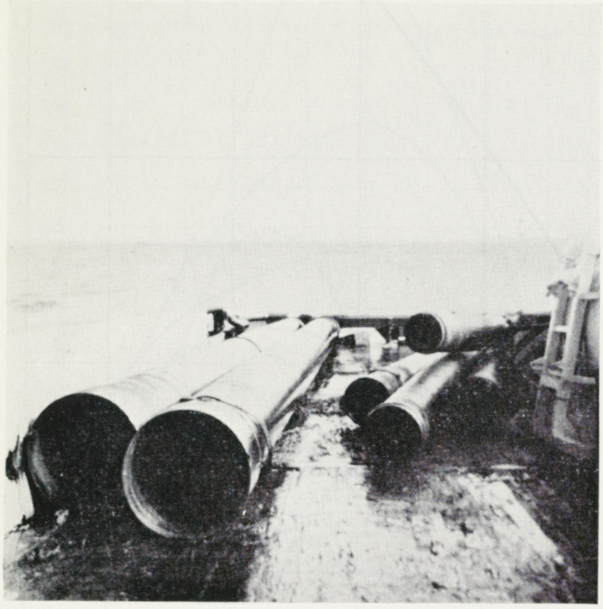


PLATE 5

Deck cargo shift at sea. The securing chains snapped during heavy weather in the North Sea. Excessive movement was prevented by container cargo on port side.

The Society in line with some other Authorities has recommended the provision of storage racks or bins in addition to inner bulwarks in order to contain cargo. If these are fitted they must be of substantial construction to avoid excessive damage and be of a design which will not hamper unloading in a seaway.

On some designs, two or more rows of portable stanchions about 1 m high and 150 mm O.D. are provided whose ends can be recessed into the cargo deck. These are particularly suitable when carrying pipe cargo and again must be of substantial construction. The distribution of deck cargo is another important factor which can effect stability and stowage in way of freeing ports, drainage areas and weather deck doors should be avoided.

TRIM

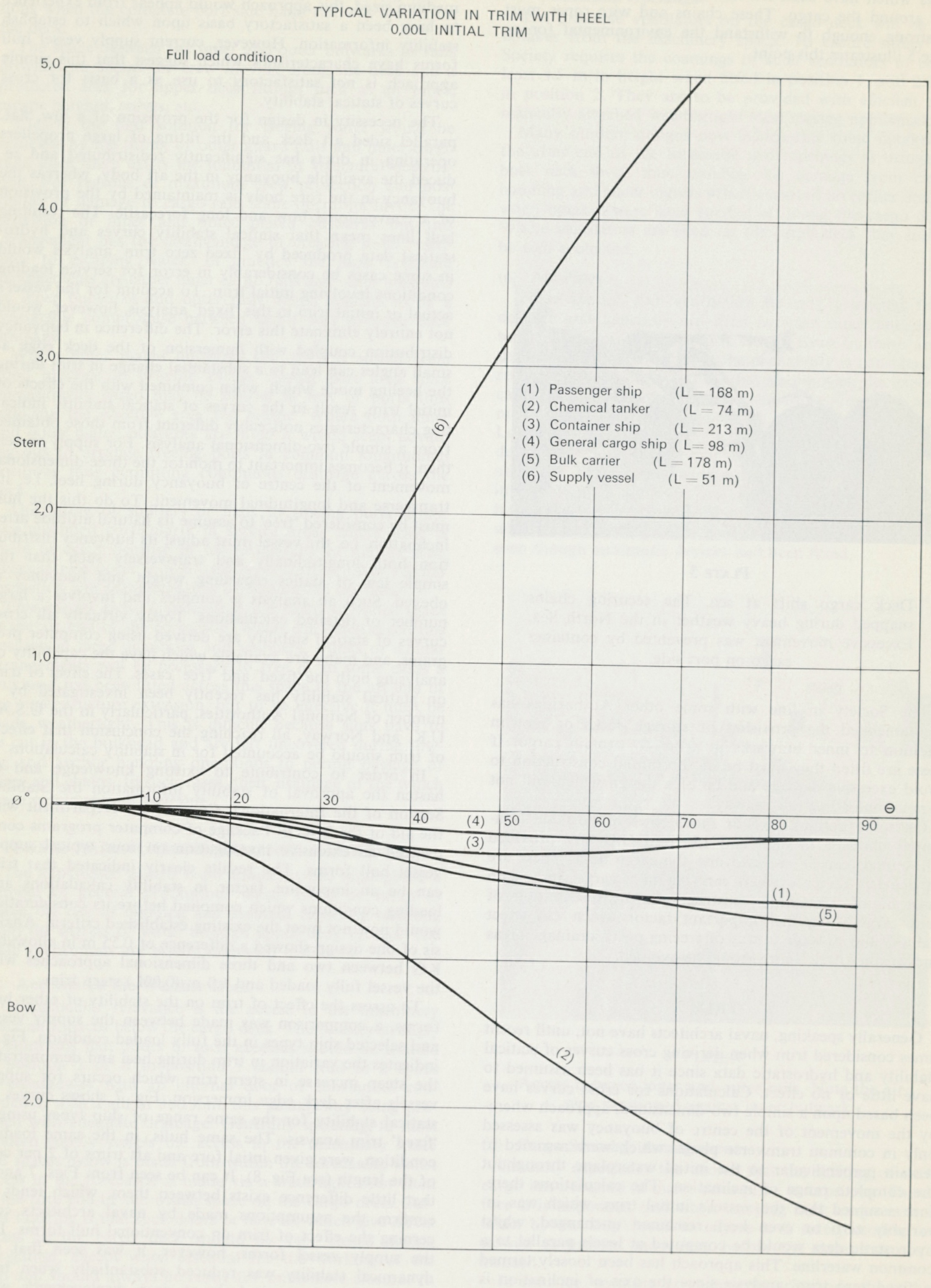
Generally speaking, naval architects have not, until recent times considered trim when deriving cross curves of statical stability and hydrostatic data since it has been assumed to have little or no effect. Calculations for cross curves have been based upon a simple two-dimensional approach whereby the movement of the centre of buoyancy was assessed only in common transverse planes which were assumed to remain perpendicular to the initial waterplane throughout the complete range of inclination. The calculations therefore assumed that the vessel's initial trim, which was invariably zero or even keel, remained unchanged, whilst hydrostatic data would be computed at levels parallel to a common waterline. This approach has been loosely termed a 'fixed zero trim' analysis since the axis of inclination is fixed both longitudinally and horizontally, rotation only occurring in a plane perpendicular to this axis.

For conventional ship forms, particularly those large and medium sized, this approach would appear from experience to have been a satisfactory basis upon which to establish stability information. However, current supply vessel hull forms have characteristics which suggest that this simple approach is not satisfactory to use as a basis for cross curves of statical stability.

The necessity in design for the provision of a low, flat, parallel sided aft deck and the fitting of large propellers operating in ducts has significantly redistributed and reduced the available buoyancy in the aft body, whereas the buoyancy in the fore body is maintained by the provision of a conventional bow and long forecastle. The resulting hull lines mean that statical stability curves and hydrostatical data produced by 'fixed zero trim' analysis would in some cases be considerably in error for service loading conditions involving initial trim. To account for the vessel's actual or initial trim in this 'fixed' analysis, however, would not entirely eliminate this error. The difference in buoyancy distribution coupled with immersion of the deck edge at small angles can lead to a substantial change in trim during the heeling mode which, when combined with the effects of initial trim, result in the curves of statical stability indicating characteristics noticeably different from those obtained from a simple two-dimensional analysis. For supply vessels then, it becomes important to monitor the three-dimensional movement of the centre of buoyancy during heel, i.e. its transverse and longitudinal movement. To do this the hull must be considered 'free' to assume its natural attitude after inclination, i.e. the vessel must adjust its buoyancy distribution both longitudinally and transversely such that the simple law of statics regarding weight and buoyancy is obeyed. Such an analysis is complex and involves a large number of detailed calculations. Today virtually all cross curves of statical stability are derived using computer programs, and many are available which have the capability of analysing both the 'fixed' and 'free' cases. The effect of trim on statical stability has recently been investigated by a number of National Authorities, particularly in the U.S.A., U.K. and Norway, all reaching the conclusion that effects of trim should be accounted for in stability calculations.

In order to contribute to existing knowledge and to hasten the approval of stability information the Stability Section of the International Conventions Department with the aid of the SIKOB-Package of computer programs commenced an extensive investigation on four typical supply vessel hull forms. The results clearly indicated that trim can be an important factor in stability calculations and loading conditions which complied before its consideration would now not meet the existing established criteria. Analysis of one design showed a difference of 0.55 m in allowable KG between two and three dimensional approaches with the vessel fully loaded and 1.0 m (0.02L) stern trim.

To assess the effect of trim on the stability of other hull forms, a comparison was made between the supply vessel and selected ship types in the fully loaded condition. Fig. 6 indicates the variation in trim during heel and demonstrates the steep increase in stern trim which occurs for supply vessels after deck edge immersion. Fig. 7 shows curves of statical stability for the same range of ship types using a 'fixed' trim anaysis. The same hulls, in the same loading condition, were given intial fore and aft trims of 2 per cent of the length (*see* Fig. 8). It can be seen from Figs. 7 and 8 that little difference exists between them, which tends to confirm the assumptions made by naval architects concerning the effect of trim on conventional hull forms. For the supply vessel forms, however, it was seen that the dynamical stability was reduced substantially when trim was considered, particularly at deep displacements with initial stern trim. Fig. 9 shows this point clearly. The effect



TYPICAL VARIATION IN TRIM WITH HEEL
0,00L INITIAL TRIM
Full load condition
5,0
4,0
3,0
Stern
2,0
1,0
Ø °
0
1,0
Bow
2,0
10
20
30
40
50
60
70
80
90
Θ
(1) Passenger ship (L = 168 m)
(2) Chemical tanker (L = 74 m)
(3) Container ship (L = 213 m)
(4) General cargo ship (L = 98 m)
(5) Bulk carrier (L = 178 m)
(6) Supply vessel (L = 51 m)
(1)
(2)
(3)
(4)
(5)
(6)

FIG. 6

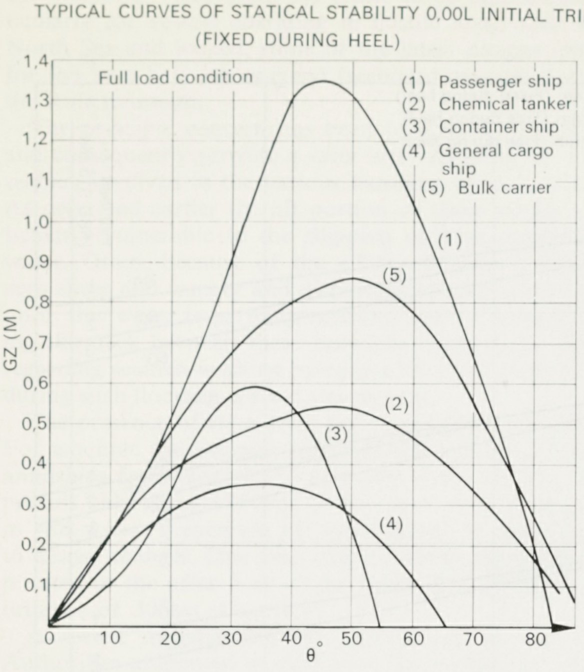


FIG. 7

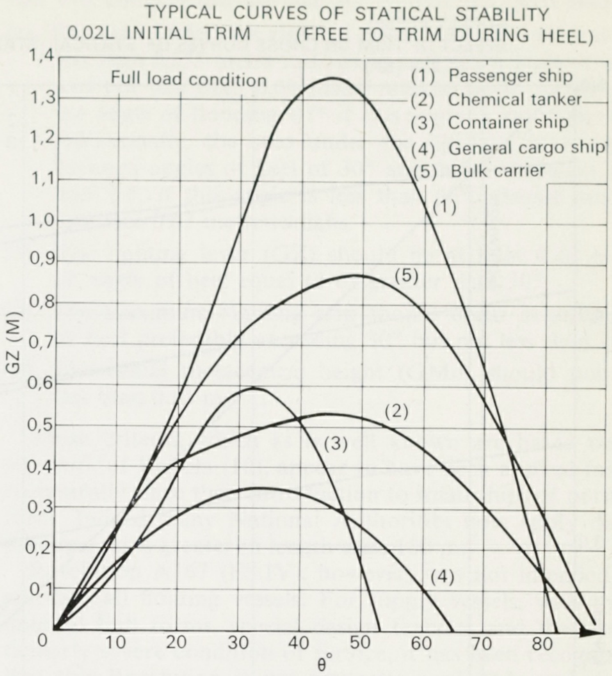


FIG. 8

Characteristic	% Change 1 ➞ 2
Area 0° ➞ 30°	—22,3
0° ➞ 40°	—30,9
30° ➞ 40°	—48,1
GZ at 30°	—41,3
θ GZ MAX	22,1° to origin
θ VAN	14,3° to origin

EFFECT OF TRIM ON THE STATICAL STABILITY CURVE OF A SUPPLY VESSEL

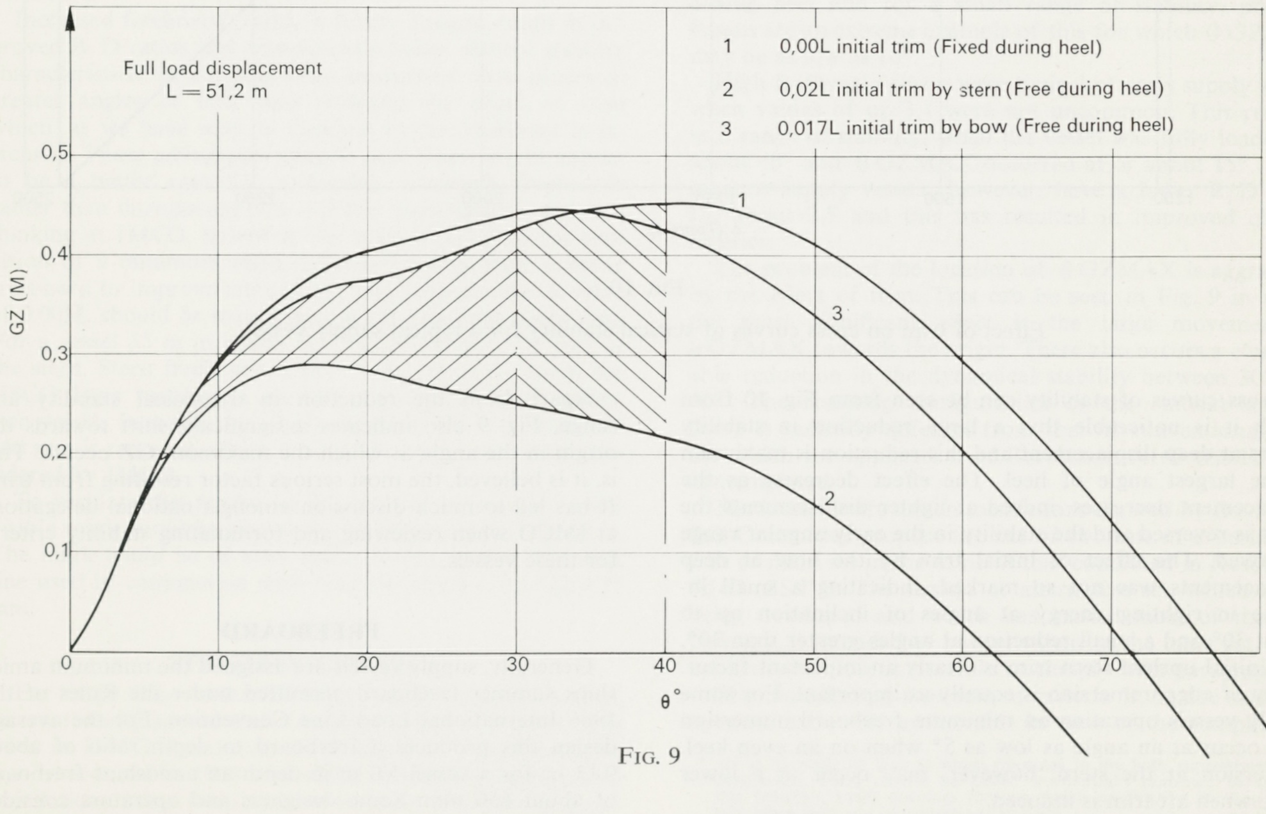


FIG. 9

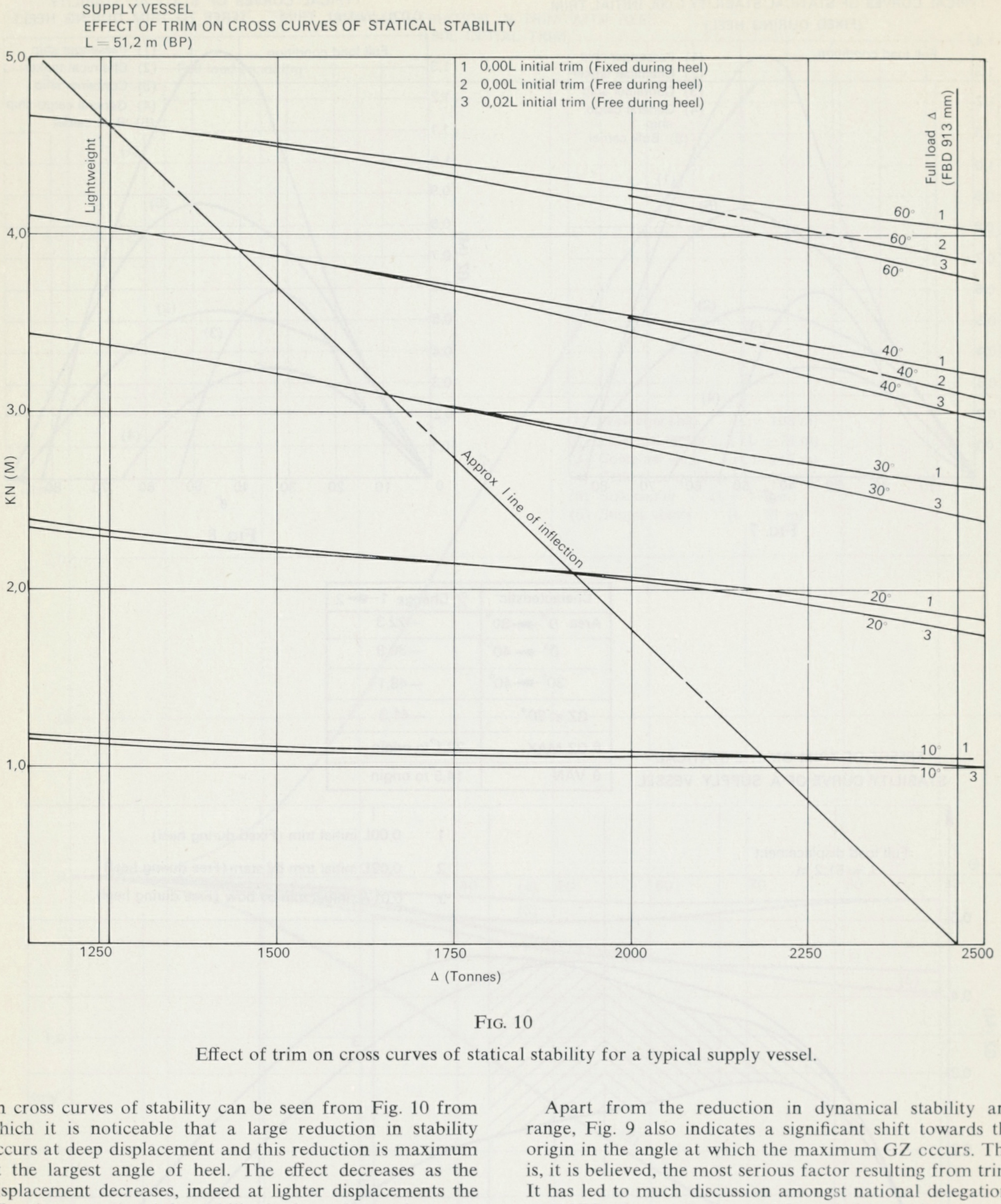


FIG. 10

Effect of trim on cross curves of statical stability for a typical supply vessel.

on cross curves of stability can be seen from Fig. 10 from which it is noticeable that a large reduction in stability occurs at deep displacement and this reduction is maximum at the largest angle of heel. The effect decreases as the displacement decreases, indeed at lighter displacements the effect is reversed and the stability in the early angular range improved. The effect of initial trim by the bow at deep displacements was not so marked, indicating a small increase in righting energy at angles of inclination up to about 30° and a small reduction at angles greater than 30°. The initial upright stern trim is clearly an important factor. Delay of edge immersion is equally as important. For some supply vessels operating on minimum freeboard immersion may occur at an angle as low as 5° when on an even keel. Immersion at the stern, however, may occur at a lower angle when aft trim is induced.

Apart from the reduction in dynamical stability and range, Fig. 9 also indicates a significant shift towards the origin in the angle at which the maximum GZ occurs. This is, it is believed, the most serious factor resulting from trim. It has led to much discussion amongst national delegations at IMCO when reviewing and formulating stability criteria for these vessels.

FREEBOARD

Generally, supply vessels are assigned the minimum amidships summer freeboard permitted under the Rules of the 1966 International Load Line Convention. For the average design this produces a freeboard to depth ratio of about 0.13 or for a vessel 5.0 m in depth an amidships freeboard of about 650 mm. Some designers and operators consider

the minimum value from the Rules to be inadequate, particularly for vessels operating in hostile areas such as the North Sea and indeed, some of the latest designs intended for this area have an assigned freeboard over twice that of the Rule minimum.

The principle concern has been to achieve a dryer ship and consequently provide a safer working platform for the seamen involved in the various handling operations at sea. As described earlier the aft portion of these vessels is particularly vulnerable to the shipping of large quantities of water. Often, because of the effect of inboard bulwarks, zero sheer and camber and dispostion of cargo over freeing ports this water is partially retained and as a result flooding the deck between these bulwarks takes place. Reports regarding seamen working momentarily waist deep in water during such flooding are not uncommon.

The presence of stern trim clearly increases the problem. For example the average design mentioned above with an amidships freeboard of 650 mm and 0.02L aft trim would possess only about 175 mm at the stern. This is, of course, in still water. Prevention of the shipping of water has led to design changes. One new class of vessel has sheer incorporated in the after 5 m of the cargo deck giving a sheer ordinate of .305 m at the A.P.

A recent investigation (9) carried out by the Dutch Authorities attempted to determine the probability of shipping water at the stern in following waves. Analysis of the results clearly demonstrated the importance of freeboard in this respect. For a supply vessel moving at 12.5 knots on an even keel in an irregular uni-directional following sea corresponding to a Beaufort Force 8 with an amidships freeboard of 750 mm, the probability of shipping water at the stern is calculated as being 37 per cent. In the same condition, with the freeboard doubled to 1500 mm, the probability decreases to only 2 per cent. Calculations in other sea states at varying speeds also show significant decreases in probability when the freeboard is increased. This investigation suggests that any assessment for water entrapment in stability calculations should be a function of freeboard.

Increased freeboard could, in future designs, result in improved B/D ratios and consequently better statical stability characteristics. In addition edge immersion then occurs at greater angles of heel thus reducing the effect of trim which, as we have seen, is lessened as the freeboard is increased. These advantages suggest that there would appear to be a better case for increasing minimum freeboards rather than introducing new stability requirements. Current thinking at IMCO, however, has been focused on the provision of a minimum stern freeboard rather than summer freeboard to improve crew safety and consider that a value of 0.005L should be maintained in all operating conditions. For a vessel 55 m in length this would provide 275 mm at the stern. Stern freeboard is not a new concept, indeed the USCG have required a minimum value for many years. For vessels between the length range 47–58 m a value of 560 mm is required, which is virtually twice that being considered by IMCO.

To regulate this freeboard in practice, presumably some visible mark could be placed on the ship's side at the stern. The mark could be of steel and a similar size as the deck line used in conjunction with load line marks, i.e. 300×25 mm.

STABILITY CRITERIA

For cargo ships under 100 m in length the only detailed, internationally agreed intact stability criteria are those outlined in IMCO Resolution A167 (ES.IV). This Resolution was introduced in 1969 and whilst containing only recommendations, it has been adopted by many National Authorities. For convenience the stability criteria are listed below:

(a) The area under the righting lever curve should not be less than 0,055 metre-radians up to $\theta = 30°$ angle of heel and not less than 0.09 metre-radians up to $\theta = 40°$ or the angle of flooding θf* if this angle is less than 40°. Additionally, the area under the righting lever curve between angles of heel of 30° and 40° or between 30° and θ f, if this angle is less than 40°, should not be less than 0.03 metre-radians.

(b) The righting lever (GZ) should be at least 0.20 m at an angle of heel equal to or greater than 30°.

(c) The maximum righting arm should occur at an angle of heel preferably exceeding 30° but not less than 25°.

(d) The initial metacentric height (GMo) should not be less than 0.15 m.

These criteria, which as is well known are based upon the work of Rahola (10), appear to have been applied fairly successfully since their introduction to small ships of normal form. Indeed many National Authorities now apply them to cargo ships greater in length than 100 m.

Resolution A167 (ES.IV), however, was not intended to embrace all floating vessels. For supply vessels, with their unusual hull forms, special design features and their particularly severe condition of service, it has been recognized that this Resolution is not generally applicable and there exists a need to develop new stability criteria relevant to this type. The formulation of new stability criteria cannot be achieved quickly since it involves a tremendous amount of research and statistical analysis and for this reason it is likely that those contained in Resolution A167 (ES.IV) will be applied to supply vessels until suitable criteria have been developed.

Compliance with these criteria in full can be extremely difficult and in some cases it may be impossible. The major difficulty is in meeting paragraph (c), i.e. the angle at which the maximum righting lever occurs (θGZMAX). This is closely related to the B/D ratio. For forms where this ratio is high, the tendency is for the θGZMAX to occur early during heel and for a small range of stability, pontoon barges are an extreme example of this for which θ GZMAX may be as low as 10°.

High B/D ratio forms were typical of early supply vessels when values of up 3.7 were not uncommon. This resulted in a range of stability, when the vessel was fully loaded, of about 40° and θGZMAX occurred at or about 15°. Many modern supply vessels, however, have a lower B/D ratio, i.e. about 2.5 and this has resulted in improved characteristics.

The problem of the location of θGZMAX is aggravated by the effect of trim. This can be seen in Fig. 9 in which the most significant effect is the large movement of θGZMAX towards the origin. There also occurs a considerable reduction in the dynamical stability between 30° and 40°. The resulting configuration of the statical stability curve is radically different from that of conventional hulls for which the application of Resolution A167 (ES.IV) was contemplated.

The latest Draft Recommendations derived by the IMCO Sub-Committee propose the use of two sets of criteria. The first or main set in the same at that contained in Resolution A167 (ES.IV). The second or alternative set, which is still being developed and which has caused most of the controversy, is intended to be equivalent to the main set, being applied when it is impossible to comply with paragraph (c).

The inclusion of two sets of criteria is, in the Author's opinion, extremely undesirable in view of the complication

* θf is an angle of heel at which openings in the hull, superstructures or deckhouses which cannot be closed weathertight immerse. In applying this criterion, small openings through which progressive flooding cannot take place need not be considered as open.

and misinterpretation which are likely during the preparation and, in particular, application of the stability information. For example, how will it be decided in practice when it is impossible to comply with paragraph (c)? There is no straightforward answer to this question. For many loading conditions which at first sight do not meet the most stringent criteria, i.e. Resolution A167 (ES.IV), it may be possible to achieve compliance by making only small changes in trim, loading distribution or displacement. In such cases it would seem unreasonable for it to be argued that compliance was an impossibility. Doubts over this point has led to suggestions that naval architects or designers would tend to adopt the less stringent of the criteria. If this did occur it would reflect the ambiguity of the IMCO requirement.

Many proposals have been suggested for these alternative criteria, however, in general there exist two main schools of thought. Both recognize the difficulty relating to θ GZMAX and have set the minimum value at 15°. They differ significantly, however, in the amount and distribution of dynamical stability required up to 30°. One school bases its criteria on the principle that to provide a level of safety equivalent to that of Resolution A167 (ES.IV) the dynamical stability should be increased as θ GZMAX is reduced. Consequently, the proposal is to increase the area under the GZ diagram linearly from 0.055 metre-radians when θ GZMAX occurs at 30° to a maximum 0.075 metre-radians when θ GZMAX occurs at 15°. This principle is consistent with the basis of the U.K. Relaxation which has been applied by the U.K. Administration for vessels of unusual characteristics (*see* Appendix III).

The second school believes the requirement to increase the dynamical stability at small angles is unreasonable, particularly in view of the need to account for water entrapment and the increase in the transverse metacetric height (GMT) with its possible undesirable effect on rolling characteristics. Their proposal is to provide 0.055 metre-radians up to θ GZMAX plus 5° provided the θ GZMAX occurs between 15° and 25°. In addition a reduction of 0.002 metre-radians is permitted per degree angle of heel the θ GZMAX is located below 25°.

The greatest difference between these two proposals occurs at θ GZMAX=15°, when the second requires only 53 per cent of the dynamical stability of the first proposal. Clearly a wide difference of opinion exists on this point.

Essential to each proposal is the location of θ GZMAX. Whilst for conventional forms this can be determined with a reasonable degree of accuracy, it is not always the case for supply vessels. The Society's investigations referred to earlier have shown that statical stability curves, particularly those at deep displacements can be of the most irregular configuration. Figs. 11, 12 and 13 give some examples which have occurred. The position of θ GZMAX in the latter two cases become questionable and could clearly lead to misinterpretation by the user unless some guidance is given by the Approval Authority.

To determine the importance of θ GZMAX as a limiting criteria an analysis was made over a wide range of loading combinations and trim on two typical hull forms using the U.K. Relaxed Criteria. It was seen that θ GZMAX was the most stringent criterion in 30 per cent of all cases, increasing in prominence as stern trim increased and generally relevant over the mean displacement range. Since supply vessels will normally operate in this range the accurate location of θ GZMAX assumes some significance. The analysis also revealed that the required area under the GZ diagram between 30° and 40° was the limiting criterion in 30 per cent of all cases and occurred principally at low displacements.

Finally, it must be said that whatever criteria are finally adopted they will be strictly related to still water which may rarely reflect the real environmental conditions experienced by the supply vessel. Some investigators have suggested taking a step nearer reality by calculating the cross curves of stability assuming the vessel supported on a wave profile. It can be shown that loss in dynamical stability with a wave crest amidships is considerable. Others see a need for a dynamic approach by mathematically relating the stability to the vessel's motion characteristics. It is unlikely, however, that these ideas will be incorporated into Regulations for many years to come until the state of the art has advanced sufficiently. In the meantime, presumably, the basic 'Rahola' approach of relating criteria to major casualty statistics will be pursued.

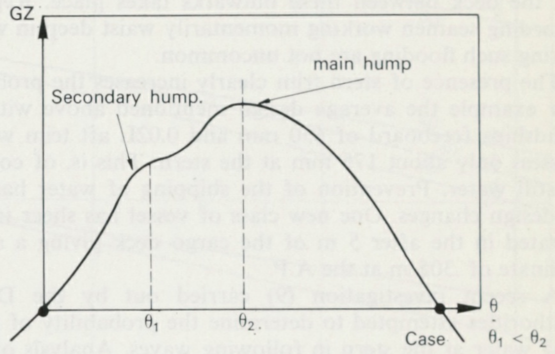


FIG. 11

Statical stability curve Case $\theta_1 < \theta_2$

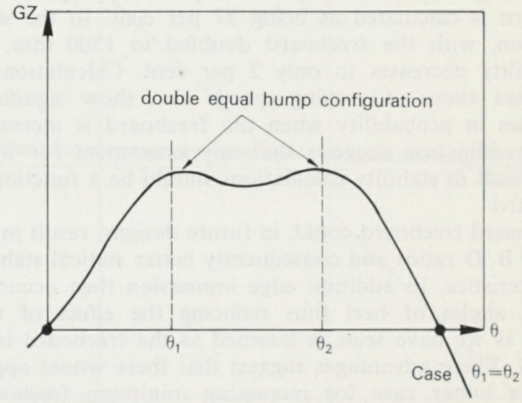


FIG. 12

Statical stability curve Case $\theta_1 = \theta_2$

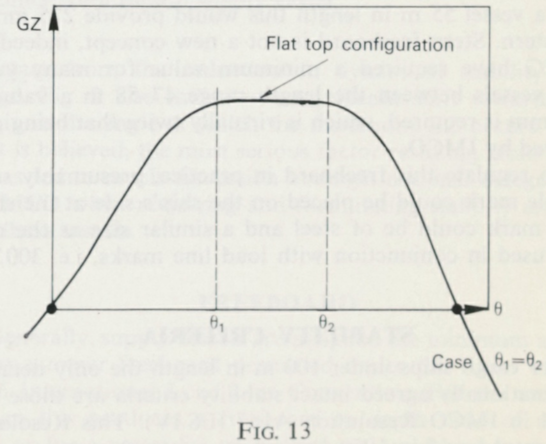


FIG. 13

Statical stability curve Case $\theta_1 = \theta_2$

STABILITY WHEN TOWING

In addition to the factors affecting stability already discussed for the vessel's supply role, there exists the danger of capsizing due to the heeling moment induced by the towline during towing operations. Whilst the Author has been unable to discover any casualty due to this particular cause between 1968–1976, it is evident that the possibility exists in practice.

Many proposals have been discussed at IMCO regarding the minimum stability criteria when towing, but opinion varies widely and for this reason it is doubtful whether any international agreement, except in operational matters will be reached in the foreseeable future.

National practices differ in detail but the general approach is to determine the heeling arm curve produced by the towline over the angular range in a specific loading condition and to superimpose this on to the statical stability diagram to derive a residual stability diagram. Specific characteristics of the latter must then comply with the required minimum criteria such that a margin against capsizing is provided. The Australian Authorities, for example, require the residual dynamical stability between 0° and 40° or θ_f, if this is less, to be at least 40 per cent of the total dynamical stability available from the statical stability diagram between the same limits. In the U.S.A. a minimum residual dynamical stability of 0.01 metre-radians has been proposed also between the same limits, whilst the Norwegian delegation suggest that the GZMAX should be at least 50 per cent in excess of the heeling arm produced by the vessel's towline pull.

The value of the towline load is an important factor in any stability calculations. For simplicity it is usually assumed that the vessel can develop its full bollard pull in a transverse plane at 90° to the centreline of ship and the heeling moment is calculated assuming this load to act at some initial angle. In the U.S.A., however, an attempt has been made to determine the towline pull in the transverse plane by estimating the proportion of the propeller thrust which can be exerted transversely when the rudders are at 45° to the centreline.

Of particular significance in any assessment is the position of towline connection to the supply vessel. Model tests carried out in the U.S.A. indicate that the danger of capsizing is a direct function of this position. The greatest danger exists when the towline is connected to only one point at or near amidships on the supply vessel while it is manœuvring. The danger is considerably reduced, however, when the towline is also connected at the stern and for this reason some National Authorities strongly recommended that a stern towing guide be used when towing. It is interesting to note that one National Authority has proposed that its special towing criteria could be disregarded if a towing guide is used. Other proposed safety measures include the provision of a quick release mechanism for use in an emergency, the restriction of the towline pull by the fitting of a constant tension winch, and some means to limit the angle of towline to the vessel's centre line since the heeling moment induced will be a direct function of this angle.

Operationally, danger exists when the vessel is towing and carrying deck cargo at the same time. This situation can lead to interference with the towline and also endanger the men working on deck. It is likely then that some restriction will be placed on the carriage of cargo during towing operations.

ICING

There also exists the danger of reduced stability due to icing and hence extra top weight on the exposed hull, superstructure sides and decks, cargo deck and deck cargo. For the majority of supply vessels this will not generally be applicable since most exploration activity has been confined to non-icing zones. Recently, however, exploration has spread further north towards, and in some cases, over the Arctic circle. Some supply vessels, with ice-breaking capability, will be operating in icing zones off the Northern, East and West coasts of Canada and as far north as the Beaufort Sea which lies to the north of Alaska. In these and similar areas allowance for icing becomes an important consideration, and should form part of the stability information placed on board the vessel.

As with towing, it is unlikely there will be any international agreement reached on the extent of allowance to be made and as a consequence national practices may vary. In the U.K. it has been the practice of the Department of Trade to apply a 'full icing allowance' or 'half icing allowance' depending on the specific location of operation, both 'full icing' and 'half icing' areas being precisely defined in terms of latitude and longitude and vary in severity. Basically the full allowance consists of applying an ice load of 30 kg/m^2 (2.93 cm in thickness) to all exposed horizontal surfaces and 15 kg/m^2 (1.47 cm in thickness) to the lateral area of one side of the vessel above the waterline which is assumed to take account of all exposed vertical surfaces. The surfaces of deck cargo are included in the allowance and to take account of rails, wires, booms, etc., the calculated weight of ice and its vertical moment are increased by 5 per cent and 10 per cent respectively. The half allowance is taken as one half of the full allowance.

The relevant weight of ice and its vertical moment would then be incorporated in the applicable loading conditions in the stability information booklet. This practice follows that given in Appendix III of IMCO Resolution A168 (ES.1V), i.e. 'Recommendations on Intact Stability of Fishing Vessels'.

From an operational point of view it would seem relevant to draw the master's attention to the information given in Resolution A269 (V111) which includes advice on how to combat the problem of ice.

WIND AND WAVES

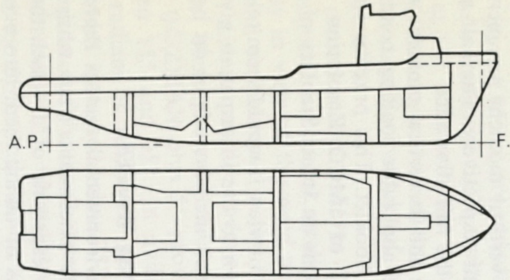
Consideration of these environmental factors represents the grey area of any stability assessment, since whilst it is obvious that they can cause capsizing of a vessel, there is at present no accurate means of taking them into account.

A number of dangerous situations could be envisaged for a vessel in a seaway under the action of wind or waves or both. The excessive motions which can occur when encountering following seas, where the length and velocity of the vessel approaches the length and velocity of the waves, is one such situation. Another, perhaps the most thoroughly investigated, is that when the vessel is subjected to beam wind and waves. It is well known that excessive rolling can occur when the beam wave period is comparable to the vessel's natural rolling period. For low freeboard vessels in beam seas, water can be shipped and trapped on deck thus inducing an additional heeling moment and a progressively increasing heeling angle. This situation may lead to the gradual capsizing of the vessel.

In the U.S.A., supply vessels must comply with a Weather Criterion (Appendix IV) in addition to the basic intact stability criteria. This attempts to limit the heel induced by a steady beam wind loading by requiring a minimum GM_T. This approach, however, tends to over simplify the problem since it ignores the presence of beam waves, which invariably would be generated, and the vessel's roll response.

More reasonable would be the theoretical approach adopted by the U.K. Administration for application to fishing vessels (8). In this case a 'dynamic heel criterion CW_r is used to demonstrate the ability of the vessel to withstand the effects of gusts, severe winds and rolling. Basically this

TYPICAL LOADING SHEET



Colour	Item	Load (Tonnes)	V.C.G. A.B. (m)	Vert. L moment (tm)	L.C.G. from A.P. (tm)	Long. L moment (tm)	Free surface Actual S.G.	Free surface Moment (tm)
Condition :— Fully loaded arrival								
Total deadweight		A						
Lightship		B						
Displacement =		A + B						
Water on deck		C						
Ice		D						
Total Displacement =		A + B + C + D	K.G.		L.C.G.		Total	
M.C.T. ('H') =			L.C.B. from A.P. ('L') =				Trim G	
Correction for free surface =				Corrected K.G. E				

Fig. 14

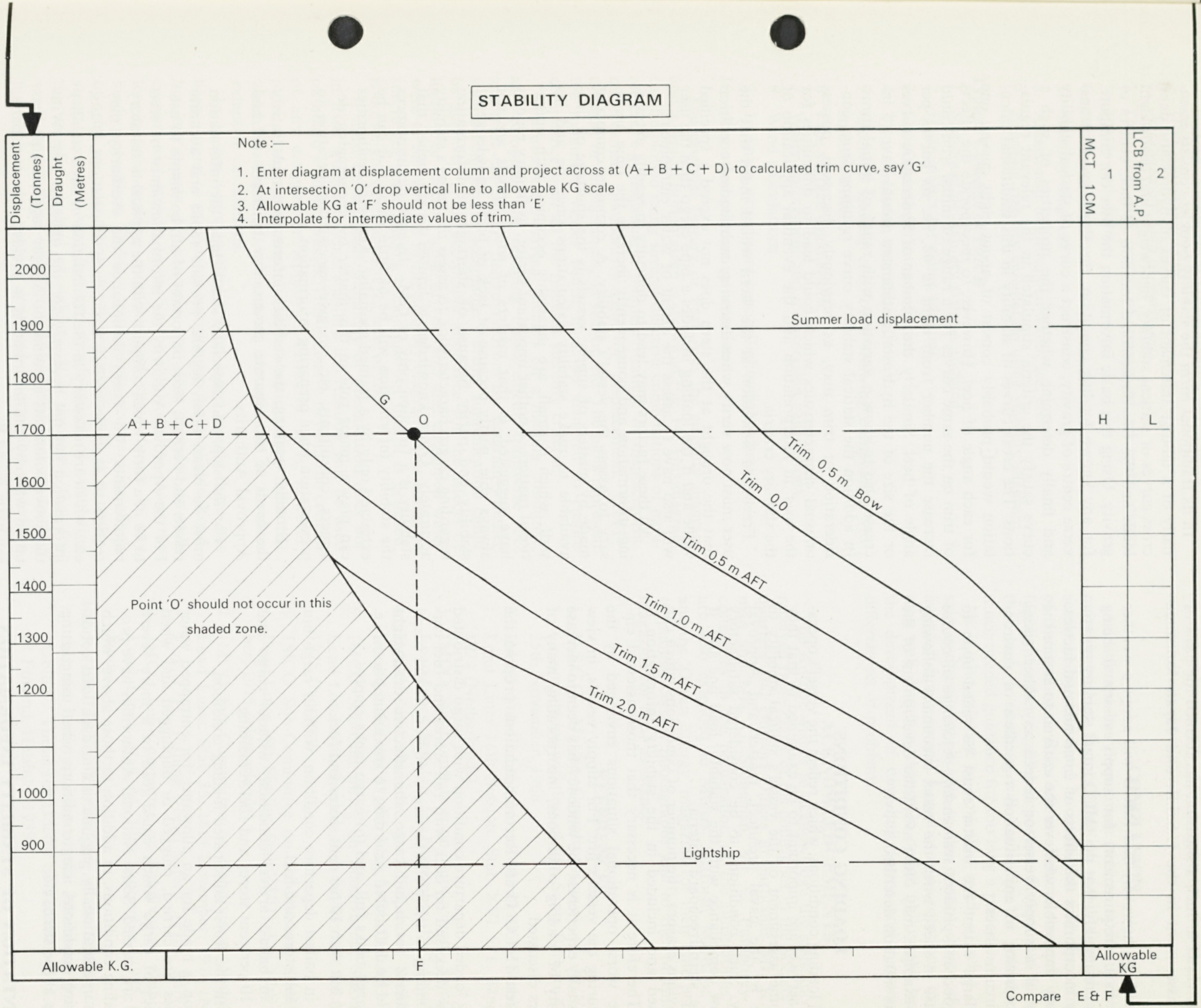


FIG. 14

Stability diagram.

approach assumes that the vessel is rolling in two dimensional irregular beam waves, when upon reaching maximum heel against the wind the vessel is suddenly subjected to a squall resulting in a dynamic heel in the direction of the wind. The vessel in such a situation has formed the basis of stability assessment in the U.S.S.R. and Japan for some time.

It is unlikely, however, that IMCO will adopt a weather criterion in its requirements for supply vessels it being assumed, presumably, that the IMCO intact stability criteria implicitly incorporates the effects of environmental factors. It is clearly impossible, whatever the criteria, to ensure the immunity of a supply vessel or indeed any other vessel against capsizing in any situation regardless of environmental circumstances.

To a large extent the master must be relied upon to display good and prudent seamanship. A change of course or decrease in speed when the vessel encounters following seas, for example, may often avoid the development of the dangerous situation described above.

LOADING CONDITIONS

Typical loading conditions, which reflect the vessel's operation and service, are important to be examined in that they enable some assessment of the vessel's statical stability to be made by the naval architect, master and Approval Authority. Such conditions to be considered satisfactory must show compliance with the stability, load line and, if applicable, stern freeboard criteria.

For obvious reasons, the number of conditions which can be assessed and included in the stability information is limited. Therefore it is necessary that those assessed embrace the worst operational situations envisaged by the owners during the vessel's life. For supply vessels, in view of the casualty experience, the tendency has been to increase the severity of loading conditions, incorporating many of the factors discussed.

The latest IMCO Draft Recommendations propose the following: —

(i) Fully loaded departure condition with cargo distributed below deck and on deck with full stores and fuel corresponding to the worst service condition. If the vessel has tanks for liquid cargo, the effective deadweight should be distributed according to two assumptions, i.e. (a) cargo tanks full, and (b) cargo tanks empty.

(ii) As (i) but with 10 per cent stores and fuel.

(iii) Ship in ballast departure condition, without cargo but with full stores and fuel.

(iv) Ship in ballast arrival condition, without cargo and with 10 per cent stores and fuel remaining.

(v) Ship in the worst anticipated loading condition.

Provision is made for an allowance for entrapped water when pipes are carried, plugged or unplugged and when other types of cargo which could lead to a more severe condition are carried. Some National Authorities have proposed loading conditions, in addition to those above, which could occur operationally, e.g. maximum deck load conditions, towing conditions, icing conditions, when using lifting appliances at the stern, etc.

PRESENTATION OF STABILITY INFORMATION

The consideration of factors such as assessment of entrapped water, possibly two sets of criteria, stern freeboard limitation and in particular, trim, in onboard stability calculations will make stability information booklets for supply vessels substantially more complex if the traditional form of presentation is used. From a user application point of view this situation is clearly undesirable and, to avoid it, new and simplified forms of presentation will have to be devised.

In its traditional form the stability booklet consists essentially of a tabular statement of curves of hydrostatic data, cross curves of statical stability, lightweight and deadweight loads and the two-dimensional location of their centres of gravity. Using this basic information the user can compute, for any known loading condition, the ships' overall transverse centre of gravity, construct a curve of statical stability and finally determine whether the properties of such a curve satisfy the criteria stipulated in the relevant regulations. The cross curves of stability in this traditional calculation would probably consist of about eight curves, one for each angle of heel. However, to incorporate the effects of trim on the same curves for a range of initial trim could increase the number from eight to 48, i.e. six curves per angle of heel. Similarly the number of hydrostatic curves or the size of the hydrostatic statement would have to increase also to show a variation with range of initial trims. In addition the statical stability curve resulting from consideration of trim may, as previously discussed, possess unusual characteristics which could lead to difficulties for the user in interpretation and the eventual application of the stability criteria.

Freeboard limitation at the stern would mean that the user must now verify when assessing each loading condition that the draught at the stern does not exceed a specified value and if the loading includes a pipe cargo then the user will also have to assess the extent of water entrapment.

All these factors may lead to difficulty in readily extracting information and consequently increase the time necessary to assess the vessel's stability. A common complaint made by masters of supply vessels is the shortage of time available to make stability calculations since he and his staff, which is small, are invariably occupied with operational matters. Another important factor, as inquiry reports have suggested, is that masters of small vessels are not always able, partly because of lack of basic knowledge of stability and partly because of shortage of information regarding cargo characteristics, to assess the vessel's stability accurately. Often knowledge is limited to the belief that stability is a function only of GM_T. In such circumstances the stability information will be of little value if it can be extracted only by a trained specialist. Approval Authorities can play a major role in this matter by encouraging designers, consultants, etc., to give special consideration to presentation when preparing information.

Broadly speaking, the stability information must satisfy the needs of two separate groups, viz. the ship's staff and Approval Authority.

For the ship, and especially small ships, the information must be clear, brief, legible, provide rapid and accurate results and perhaps most important of all, be simple to use. For the Approval Authority the information must be seen to be in a form which complies with the applicable statutory requirements. To assess compliance the Authority may require to examine certain basic data, calculations, plans, etc., which are necessary in the formulation of such stability information but not necessary for the onboard user. With the above comments in mind the Author suggests a presentation whereby the information is divided into two parts, say parts A and B.

Part A would be intended principally for the onboard user and contain only simplified information, whilst part B would contain more detailed data, calculations, etc., which form the basis of the simplified information and would generally be of interest only to an Approval Authority.

Typical Contents of Part A

1. General particulars.
2. Special notes to the master regarding stability (free surface, stabilization tanks, freeing ports, etc.).
3. Unit conversion table.
4. Arrangement plan of tanks, etc.
5. Capacities and centre of gravity of all tanks, cargo spaces and stores.
6. Deadweight scale.
7. Stability diagram.
8. Assessment of entrapped water, ice, etc.
9. Example stability calculation using stability diagram.

Typical Contents of Part B

1. Statement of stability criteria adopted.
2. Hydrostatics for range of trims.
3. Cross curves for range of trims.
4. Trim limitation calculations.
5. Specified loading conditions showing compliance with stability criteria.
6. Free surface calculations.
7. Water entrapment and ice accretion calculations.
8. Wind calculations.
9. Inclining experiment report/lightweight check.
10. Any other relevant information.

The essential feature of Part A is the stability diagram and a typical presentation is shown in Fig. 14. This diagram can be plotted using several different combinations of parameters, e.g. deadweight moment/draught or displacement, displacement moment/draught or displacement, GMT/draught or displacement, as shown, etc. The plots represent the minimum values which must be achieved to ensure compliance with all the relevant stability criteria.

The user then, having decided on a loading condition, and computed the resulting displacement, transverse KG and trim can quickly determine from the diagram whether such a condition is acceptable. Further simplification could be achieved if an acceptable allowance for the effects of liquid-free surface, icing (if relevant) and entrapped water were to be included in the stability diagram. This could be severe for some loading conditions but this, it is believed, is outweighed by the need for ease of use.

It must be said that simplicity of presentation does not in itself guarantee that stability information will be used on board a supply vessel. This is clearly beyond the control of any designer or Approval Authority. It is, however, more likely to be used if made simple rather than that which is unwieldy, voluminous, complicated or involves extensive calculations.

In recent years particularly on large vessels there has been a trend to provide on board computational aids. Loading instruments, for example, are often provided on bulk carriers and tankers to enable the ship's staff to assess the longitudinal bending moments and shear forces in any loading condition. Stability instruments incorporating effects of trim are now being developed which will, it is understood, provide a very fast and simple means of calculating the vessel's standard of stability. It is envisaged that for small vessels these could provide a useful back-up to, but not replace, the normal approved information placed on board. Indeed, this could be the most acceptable solution to the problem of onboard calculations in small vessels, but clearly this idea will have to be approved by National Authorities. Further, owners will need convincing of the operational necessity for computational aids before any expenditure is sanctioned.

CONCLUSION

An attempt has been made here to review some of the more important factors affecting stability which are presently being considered in connection with the formulation of International Regulations. Future research may well highlight other equally important factors, but it is hoped that such a situation will not delay the introduction of much needed guidance for too long a period.

It is also relevant to note that much of what has been said is directly applicable to the present design characteristics of supply vessels. Logistics and operational experience may well dictate a change in design philosophy leading to the development of new hull forms and consequently a different set of problems.

ACKNOWLEDGEMENTS

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APPENDIX I

Incidence of Collision/Contact Damage.

Comparison between ship types, the time periods and other causes of damage.

Type of ship	Period: 1–66 to 3–74 cause: other			Period: 4–74 to 5–76 cause: other			Period: 44–74 to 5–76 cause: coll. or contact		
	A	B	C	A	B	C	A	B	C
Supply ships	28	206	13.6	61	219	27.9	73	219	33.3
Small tankers	39	270	14.4	19	102	18.6	13	102	12.7
Small dry cargo	58	496	11.7	29	241	12.0	35	241	14.5
Fishing vessels	70	1010	6.9	52	589	8.8	33	589	5.6
Tugs	38	1224	3.1	23	708	3.2	42	708	5.9
Ferries	21	560	3.8	4	153	2.6	6	153	3.9

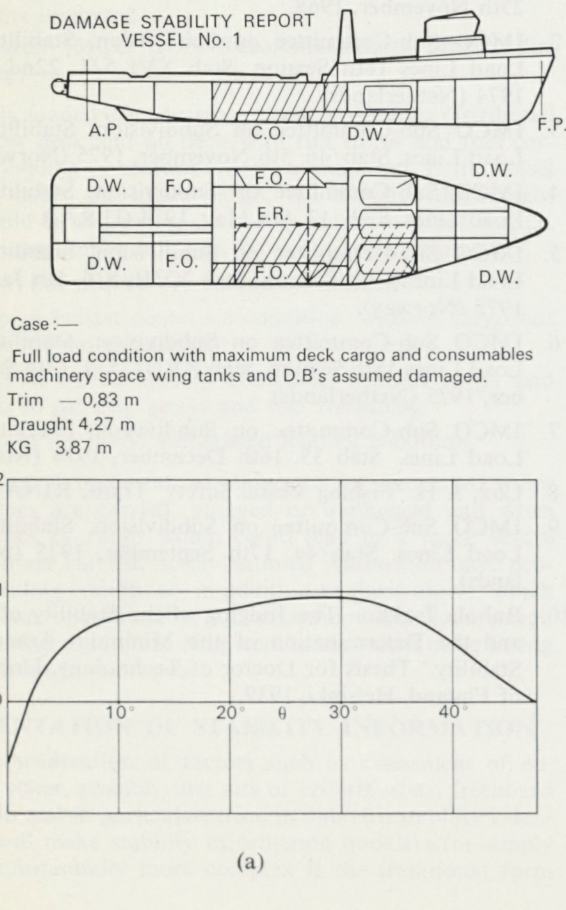
NOTE: A = number of reported damage cases.

B = total years' service.

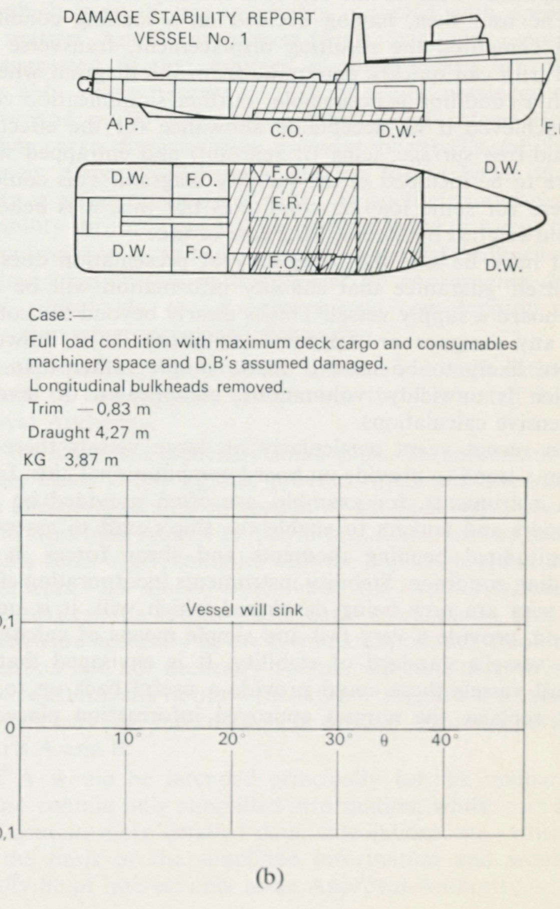
C = incidence per 100 years' service

$$\text{Incidence of damage} = \frac{\text{No. of damage cases}}{\text{Aggregate years in service}}$$

APPENDIX II

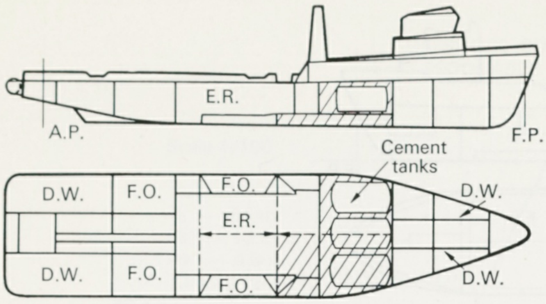


(a)



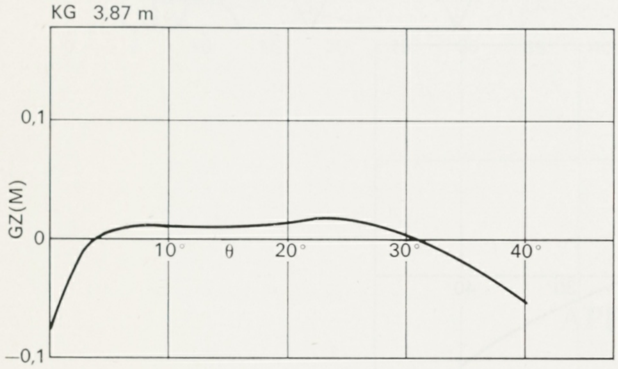
(b)

DAMAGE STABILITY REPORT
VESSEL No. 1



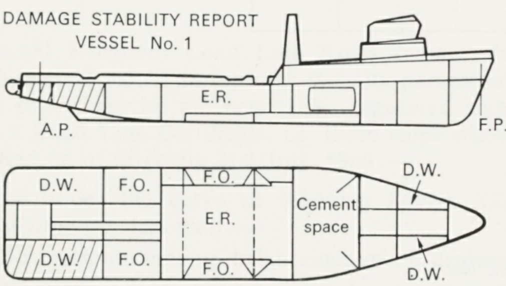
Case :—

Full load condition with maximum deck cargo and consumables
Cement space and D.B's assumed damaged
Trim — 0,83 m
Draught 4,27 m
KG 3,87 m



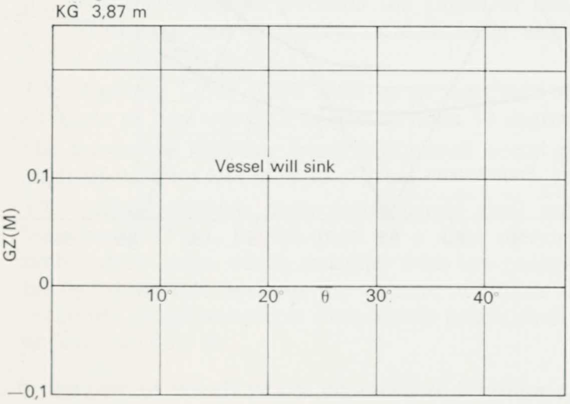
(c)

DAMAGE STABILITY REPORT
VESSEL No. 1



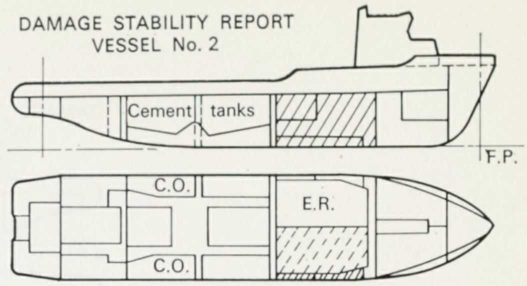
Case :—

Full load condition with maximum deck cargo and consumables
Aft D.W. tank assumed damaged
Trim – 0,83 m
Draught 4,27 m
KG 3,87 m



(d)

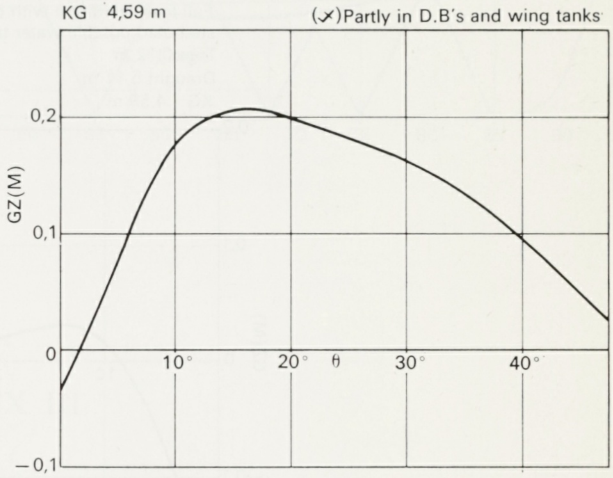
DAMAGE STABILITY REPORT
VESSEL No. 2



Case :—

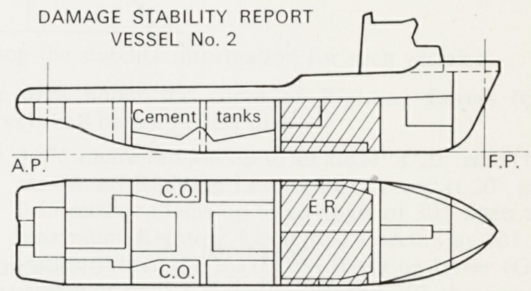
Full load condition with deck cargo and consumables (✓)
machinery space D.B's and wing tanks assumed damaged.
Trim 0,12 m
Draught 5,11 m
KG 4,59 m

(✓)Partly in D.B's and wing tanks



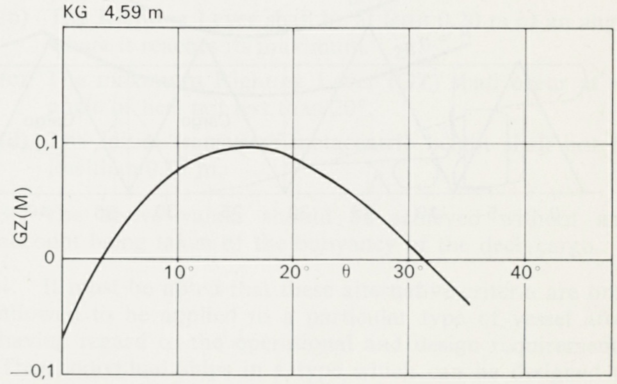
(e)

DAMAGE STABILITY REPORT
VESSEL No. 2



Case :—

Full load condition with deck cargo and consumables
machinery space assumed damaged. Longitudinal
bulkheads removed.
Trim 0,12 m
Draught 5,11 m
KG 4,59 m



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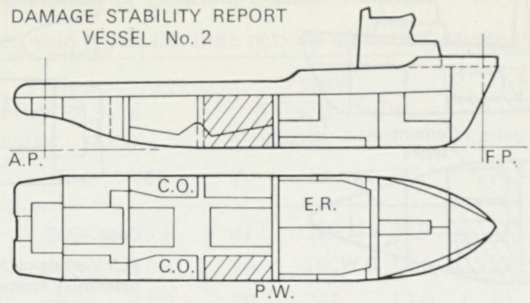
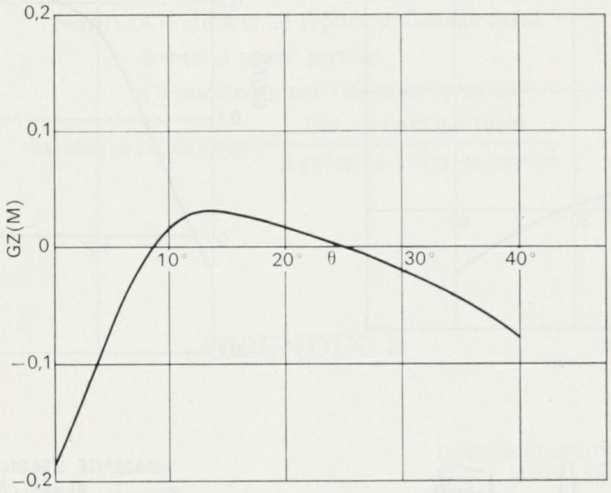
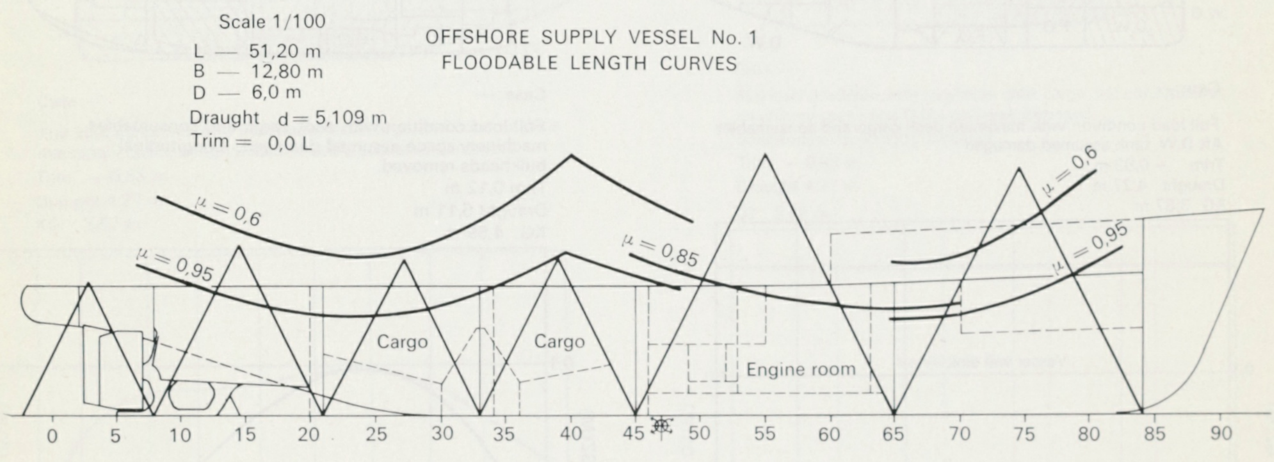


Case:—

Full load condition with deck cargo and consumables starboard potable water tank assumed damaged.
Trim 0,12 m
Draught 5,11 m
KG 4,59 m

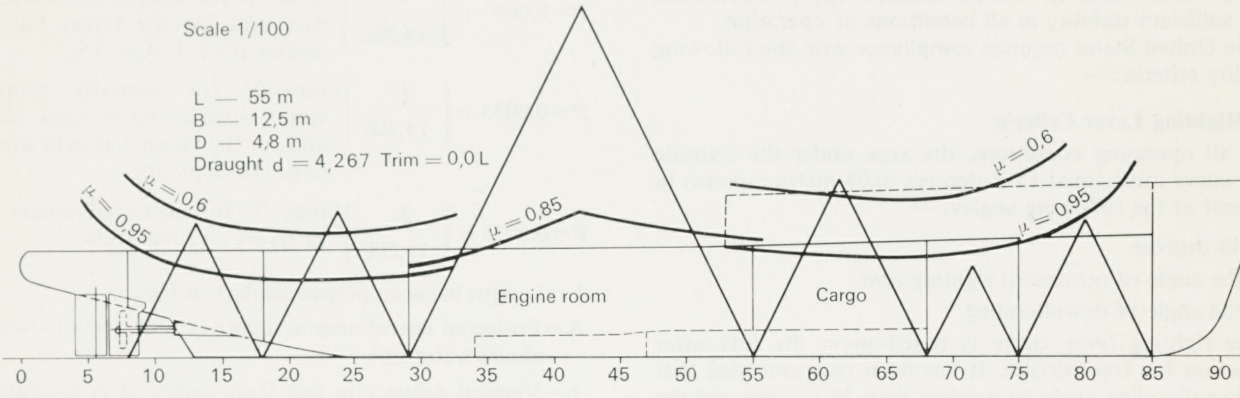


(g)



(h)

OFFSHORE SUPPLY VESSEL No. 2
FLOODABLE LENGTH CURVE



(i)

APPENDIX III

Deviation from the requirements of the U.K. Load Line Rules which may be permitted, subject to approval by the Administration.

1. The United Kingdom Load Line Rules require the following numerical values for specific stability parameters which have to be met by all 'new' U.K. registered ships issued with a Load Line Certificate, i.e. those ships whose keels were laid on or after the 21st July, 1968: —

(a) The area under the curve of Righting Levers (GZ curve) shall not be less than: —

- (i) 0.055 metre-radians up to an angle of 30 degrees;
- (ii) 0.09 metre-radians up to an angle of either 40 degrees or the angle at which the lower edges of any openings in the hull, superstructures or deck-houses being openings which cannot be closed weathertight, are immersed if that angle be less;
- (iii) 0.03 metre-radians between the angles of heel of 30 degrees and 40 degrees or such lesser angle as is referred to in (ii).

(b) The Righting Lever (GZ) shall be at least 0.20 m at an angle of heel equal to or greater than 30 degrees.

(c) The maximum Righting Lever (GZ) shall occur at an angle of heel not less than 30 degrees.

(d) The initial transverse metacentric height shall not be less than 0.15 m. In the case of a ship carrying a timber deck cargo which complies with sub-paragraph (a) by taking into account the volume of timber deck cargo the initial transverse metacentric height shall not be less than 0.05 m.

2. In the case of vessels whose characteristics render compliance with paragraph 1(c) impossible the following criteria can be accepted in place of that given in paragraph 1 when examining the stability information for such vessels: —

(a) The area under the curve of Righting Levers (GZ curve) shall be not be less than: —

- (i) 0.075 metre-radians up to an angle of 20° when the maximum Righting Lever (GZ) occurs at 20° and 0.55 metre-radians up to an angle of 30° when the maximum Righting Lever (GZ) occurs at 30° or above. Where the maximum Righting Lever (GZ) occurs at angles between 20° and 30° the corresponding requisite area under the Righting Lever Curve shall be determined by linear interpolation.
- (ii) 0.03 metre-radians between the angle of heel of 30° and 40° or such lesser angle as is referred to in paragraph 1 (i.e. angle of flooding).

(b) The Righting Lever shall be at least 0.20 m at an angle where it reaches its maximum.

(c) The maximum Righting Lever (GZ) shall occur at an angle of heel not less than 20°.

(d) The initial transverse metacentric height shall not be less than 0.15 m.

3. The above values should be achieved without any account being taken of the buoyancy of the deck cargo.

4. It must be noted that these alternative criteria are only allowed to be applied to a particular type of vessel after having regard to the operational and design requirements. Thus individual ships in a type which can be designed to meet the provisions of paragraph 1 should not be approved against the alternative given in paragraph 2.

APPENDIX IV

United States practice regarding stability of offshore supply vessels.

In order to be considered satisfactory for Load Line assignment in ocean service, all offshore supply vessels must have sufficient stability in all conditions of operation.

The United States requires compliance with the following stability criteria: —

(a) **Righting Lever Criteria**

In all operating conditions, the area under the righting lever curve must equal 15 ft-degrees (0.08 metre-radians) to the least of the following angles: —

(i) 40 degrees,

(ii) the angle of maximum righting arm,

(iii) the angle of downflooding.

The righting lever curve is based upon the KG after correction for free surface. It has been recommended that the downflooding angle be not less than 25 degrees and the angle of maximum righting arm be not less than 15 degrees.

(b) **Weather Criteria**

In all operating conditions, the minimum required GM determined from the following formula must also be met: —

$$GM = \frac{PAh}{\triangle \tan \theta}$$

Where:

$P = 0.005 + \left[\frac{L}{14{,}200}\right]^2$ tons/ft^2 for oceans, coastwise service and for the Great Lakes in winter (Oct. 1–Apr. 15).

$P = 0.0033 + \left[\frac{L}{14{,}200}\right]^2$ tons/ft^2 for partially protected waters such as lakes, bays, sounds and for the Great Lakes in summer (Apr. 16–Sept. 30).

$P = 0.0025 + \left[\frac{L}{14{,}200}\right]^2$ tons/ft^2 for protected waters such as rivers and harbours.

L = Length between perpendiculars in feet.

A = Projected lateral area in square feet of portion of vessel above water line.

h = Vertical distance in feet from centre of A to centre of underwater lateral area or approximately one-half draft point.

$\triangle$ = Displacement in long tons.

θ = Angle of heel to one-half the freeboard to the deck edge or 14 degrees whichever is less.

The weather criteria is the basic stability criteria which the United States administration applies to all vessels.

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In all operating conditions, the area under the righting lever curve must equal 15 ft-degrees (0.08 metre-radians) to the least of the following angles: —

(a) 40 degrees.

(b) the angle of maximum righting arm.

(c) the angle of downflooding.

The righting lever curve is based upon the KG after correction for free surface. It has been recommended that the downflooding angle be not less than 25 degrees and the angle of maximum righting arm be not less than 15 degrees.

(ii) Weather Criteria

In all operating conditions, the minimum required GM determined from the following formula must also be met: —

$$GM = \frac{PAh}{\Delta \tan \theta}$$

Where:

$P = 0.005 + \left[\frac{L}{14{,}200}\right]$ tons/ft² for oceans, coastwise service and for the Great Lakes in winter (Oct. 1–Apr. 15).

$P = 0.0033 + \left[\frac{L}{14{,}200}\right]$ tons/ft² for partially protected waters such as bays, local sounds and for the Great Lakes in summer (Apr. 16–Sept. 30).

$P = 0.0025 + \left[\frac{L}{14{,}200}\right]$ tons/ft² for protected waters such as rivers and harbours.

L = Length between perpendiculars in feet.

A = Projected lateral area in square feet of portion of vessel above water line.

h = Vertical distance in feet, from centre of A to centre of underwater lateral area or approximately one-half draft point.

Δ = Displacement in long tons.

θ = Angle of heel to one-half the freeboard to the deck edge or 14 degrees whichever is less.

The weather criteria is the basic stability criteria which the United States administration applies to all vessels.

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Discussion

on

Mr. D. T. Boltwood's Paper

SOME FACTORS AFFECTING THE STABILITY OF OFFSHORE SUPPLY VESSELS

Paper No. 3. Session 1976-77

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Hon. Sec. J. C. Huskins
71, Fenchurch Street, London, EC3M 4BS

Discussion on Mr. D. T. Boltwood's Paper

SOME FACTORS AFFECTING THE STABILITY OF OFFSHORE SUPPLY VESSELS

Mr. J. McCallum

I much enjoyed listening to the presentation of this paper, one which I believe to have been overdue in view of the onerous operational duties these vessels perform, and now that it has come, it is clear that Mr. Boltwood has made an authoritative job of it.

His remarks about closing appliances for deck openings emphasize the prime importance of operational security in these vessels which are subject to much water on deck.

The flat swim-ended stern referred to in the section on trim will generate a considerable negative pressure at speed, resulting in a rapid change of trim by the stern, clearly undesirable from the aspect of stability and stern wash, particularly with a history of swamping from the rear in stern seas. Can any useful change be made in the under-water form aft?

Congratulations to the Author on a splendid paper. I hope he will not feel inhibited from presenting something similar in a more public forum.

Mr. L. Beckwith

First of all I must congratulate Mr Boltwood on producing such an extremely readable paper, which, I am sure, will be read with great interest by many colleagues and, in particular, those in the field, who have in recent years had to try to explain the problems of supply ship stability to builders and owners.

It is a long time since we had a paper on stability presented to the Technical Association; in fact, we need to go back as far as Session 1925-26 when W. Watt read a paper entitled 'Stability and Seaworthiness of a Collier'.

This paper, although relating to a special type of ship, highlights the fact that the Society is becoming more and more involved in stability matters, in its role as the agent of national administrations, in connection with the issue of 1960 SOLAS and 1966 Load Line Certificates. It is important, therefore, that we must be aware of the latest developments in this field, particularly in view of the problems which are frequently arising with the development of more specialized types of ship.

Chapter B.102 (1976 Rules) of the Society's Rules states: 'The Rules do not cover certain technical characteristics, such as stability, trim, hull vibration, etc. The Committee cannot assume responsibility for these matters but is willing to advise upon them on request.'

B.105 of the Rules, however, gives draught and stability requirements for specialized ships, viz. 'In the cases of offshore supply ships, offshore tug/supply ships, dredgers, hopper dredgers, sand carriers, hopper barges, or reclamation craft, each ship proceeding to sea is to comply with the draught and stability requirements of the National Authority and is to have on board sufficient stability data to enable it to be properly loaded and handled or, where appropriate, for the ship to be properly towed.'

B.103 (1976 Rules) has, in fact, only appeared in the Rules within the last few years, but gives an indication of how the Classification Committee now see the importance of stability, and it is my opinion that it may not be too long before this paragraph is extended to other types of ship.

With regard to the paper, I only intend making a few points without going into great detail.

Considering the chapter on casualties given in page 5 of the paper, I think it is important to point out that in no case could inadequate stability be found as a primary cause of the casualty, but it is obvious that stability was impaired by external factors.

When considering the stability of the offshore supply ship, its inherent characteristics and function must be taken into account as well as the various factors involved, such as water on deck, water entrapped in cargo, freeboard and trim. Mr Boltwood has covered these effects in some detail in the paper and goes on to discuss the criteria now being used to evaluate the stability of these ships.

Mention is made of IMCO Resolution A167 (ES.IV) and the stability criteria contained therein which were based largely on the work of Rahola, and in his paper Mr. Boltwood states that they implicitly incorporate the effects of environmental factors. It must, however, be pointed out that these criteria are based on a quasi/pseudo dynamic approach which used very simplified assumptions.

Until about 10 years ago the art of stability had not progressed very far beyond Rahola, and in many ways it could have been compared with the situation regarding the strength of ships some 20-25 years ago. Tremendous strides have been made in the study of ships' structures in the past two decades but developments in the study of stability have taken place at a much slower pace.

'The International Conference on Stability of Ships and Offshore Vehicles, which took place at the University of Strathclyde in March 1975, did, however, provide an opportunity to exchange ideas and give information on research and development in this field. The conference showed that a considerable amount of work is being carried out in various parts of the world, but that there is a need for a correlation of these efforts, with a view to developing new criteria which will take into account all the various factors involved. There is still a long way to go but it is anticipated that eventually this research will produce far more logical and practicable criteria which can be applied to all types of ship. It is hoped that the Society will keep up to date with these developments through membership of working groups which are actively engaged in studying the problems.

Referring to the proposed legislation for the stability of supply ships, it seems that the problems of damage stability for existing ships would, in many cases, be more difficult to resolve than those for new ships. With so many supply ships already in service, has Mr. Boltwood given any thought to alternative means of achieving the same end result? One possibility which comes to mind would be requirements for special resilient fenders (e.g. inflatable fenders), which could be carried by supply ships.

Mr. M. B. Leese

Mr. Boltwood is to be congratulated on an extremely readable paper and a very colourful presentation. Bearing in mind his remarks about water entrapment on deck, I would like to ask the following:

Considering that the watertight integrity of the cargo space would minimize the retention of water on deck, would it not be possible as a design concept to increase the length of the average supply vessel by, say, 50 per cent and completely 'box in' the cargo area with flush sides and a

conventional cover folding aft? The well thus created would be the depth of the highest anticipated cargo stow and as long as the longest pipes likely to be carried, and would constitute essentially an intact space. The reason for specifying flush sides is that cargo could be simply dragged out without fouling on projections or under any overhangs.

Obviously, when cargo is being worked at the rig, the covers would be off the cargo area and the serious possibility of an ingress of water would exist. However, this possibility could be reduced by having a coaming around the cargo area of certainly not less than about 3 metres. Any water finding its way over the coaming would be drained by large diameter scuppers at deck level with non-return devices, or, alternatively, by pumps which trip automatically when the cargo covers are folded back.

So that the anchor-handling function is not lost, an extension of the cargo deck abaft the enclosed cargo area would house the retrieving winch.

Mr. R. J. BELL

My first point is in respect of the plugging of pipes to reduce the effect of entrapped water when carried as deck cargo. The Author has found that, in practice, the blanking of pipes is not generally carried out and is not thought feasible unless dealt with at the pipe manufacturing plant. However, the specification issued by the American Petroleum Institute covering the manufacture and supply of casing pipe states that protective covers must be fitted to the threads at both ends of the pipe prior to despatch from the works. It would appear to be a relatively simple job to modify these thread protectors to act as watertight plugs which need not be removed until the casings are required at the drilling platform.

My second point is a general question regarding the level of technology used in the assessment of the stability of ships. The Author implies in his paper that no real advance has been made in the practical estimation of stability and that the stability criteria as recommended by IMCO, based on work done by Rahola in the 1930's, could benefit from revision in the light of subsequent work. What can we, as Surveyors applying the stability regulations, do to influence this situation?

Finally, my thanks to the Author for shedding light on a number of interesting facets of supply vessel design and operation.

Mr. G. A. SMITH

May I add my congratulations to those of the previous contributors on Mr. Boltwood's excellent paper.

First, I would like to take up the point made in the discussion regarding sea fastenings for the deck cargo. Whilst the Society is not automatically involved in the inspection of sea fastenings, occasions have arisen where a contractor has requested our services, as an independent authority, to issue a certificate relating to the satisfactory arrangement of sea fastenings for a particular 'load out', and this has been done. However, it is obvious from Mr. Boltwood's paper that deck loads on supply vessels, particularly in the North Sea, can experience very high accelerations and impose heavy loads on the fastenings. For particularly large individual loads it is quite likely that the contractor has sufficient data available to be able to arrive at a quantitative solution to the sea fastening problem and in these cases, if certification is required, it is possible for the calculations and arrangements to be appraised before any inspection is carried out. For most 'load outs', however, the Surveyors have little or no data available and must rely on the qualitative assessment—if it looks good, it is good. To date there appears to be very little feed-back regarding shifting of deck cargoes on supply vessels which the Outport Surveyor can use. Consequently, the first step in assessing any loads imposed on the fastenings would be to have some knowledge of the probable accelerations involved. Perhaps Mr. Boltwood could comment on this.

My second point is the question of freeboard, the importance of which has already been emphasized. The empirical formulæ used in the Load Line Conventions were developed by considering the conventional type of ship of 15 to 20 years ago. Such formulæ are probably quite satisfactory when applied to the type of vessel for which they were envisaged but are difficult to apply to a vessel such as the supply ship where the configuration differs radically from the conventional. Indeed, the geometrical manipulations that are performed to arrive at a minimum freeboard for unconventional type vessels, owe more to the theories of Euclid than to those of Archimedes. I was, therefore, interested to note the comments on page 15 of the paper regarding investigations, made by Dutch authorities, relating freeboard to the probability of deck wetness.

The four basic factors affecting freeboard are: structural strength, stability, survivability and deck wetness. Two others may be considered, i.e. protection of crew and the prevention of the ingress of water through exposed openings, but these are themselves related to the other four and particularly to deck wetness. Over the past few years, as the tools have become available, great strides have been made in the fields of structural strength, stability and survivability in moving away from the empirical approach towards a more general and flexible approach but, as yet, there appears to have been little work carried out on the general approach to the deck wetness problem. To persist in the empirical, such as the provision of a minimum stern freeboard related to a single parameter, as being considered by IMCO, is to maintain an inflexible 'stop-gap' situation and does not solve the underlying problems. The relating of freeboard to a probability of deck wetness, considering the vessel as a whole and using the kind of ship response techniques developed in the structural field, seems to provide a useful way of tackling the problem and I would value Mr Boltwood's comments on this.

Capt. V. ANDERSON

I have read Mr. Boltwood's paper with interest and the remarks I am about to make are not in the nature of criticism—they are observations which have some bearing on the subject under discussion and will, I hope, be the means of helping the people on these supply ships.

Before reading this paper and before seeing the slides taken by Mr. Boltwood, I had seen photographs of cargo on decks of supply ships lashed by chain and it seemed to me that the chain, in most cases, was very light and too small for the job it had to do.

The idea of using stanchions to prevent movement of cargo is an excellent one but it does need to be backed up by lashings and, equally, as in the majority of cases where no stanchions are fitted, the lashings must be strong enough for the job they have to do and must be set up tight. Once the cargo moves stability is seriously affected.

Mr. J. CRAWFORD

Mr. Boltwood is to be congratulated on two counts, firstly on his presentation of a very interesting paper and secondly, in the course of obtaining material for his paper, of having experienced the delights or otherwise of actually sailing in an offshore supply ship in the North Sea, an experience which I am sure will have made him fully conscious of 'stability' in regard to such ships.

Offshore supply ships of various shapes and sizes have

been dealt with by MDAPAD in respect of machinery and piping systems, the arrangements having been considered on the basis of cargo ships.

It is rather disturbing, though not surprising having regard to the extreme conditions under which these ships operate, to read of such a heavy casualty list, i.e. 32 ships lost over a period of eight years and in particular one-third of which have been lost due to flooding of various compartments due to collision with the support structure of the rig/platform.

It is pleasing to note that some national authorities are considering taking action regarding the safety of these ships. Could the Author give any indication of similar action by the classification societies.

Two points which I am sure will be of interest to the 'Offshore Services Department' are the apparent regularity with which these ships make contact with the structure of the rigs and also of the 'snatch lift' method of unloading cargo, having regard to the hazard involved to personnel.

Just one more point. A ship is a ship not a vessel, e.g. Offshore Supply Ship.

WRITTEN CONTRIBUTION

Mr. O. M. Clemmetsen

I was very interested to read this paper, with the frequent references to the adverse effects of having a very low freeboard for the after deck in respect of both stability and safety of working.

In an article which appeared in Lloyd's List in May, 1973 there was a full description of how the supply ships operate in the vicinity of rigs. I wrote to its author at that time that I had recently been on a trial trip of one such vessel in Denmark when it was apparent that, with the ship loaded, the after deck could be constantly awash in any sort of seaway, and asked what were the reasons for the present design being perpetuated, considering that the various structures being supplied were well above sea level, and therefore a low freeboard on the supply vessel appeared to be irrelevant. I did not receive a reply to my question, but it may be that at that time supply ships were supposed to be capable of towing as well as anchor handling, whereas the tendency lately is to produce different designs of ships for specialized tasks.

However, I still think it would be practicable to extend the forward superstructure all fore and aft as in a shelter deck or, if more convenient for shipping anchors, that such a superstructure extend to a distance from the stern sufficient to allow anchor stowage on a short after deck near the water line. A extension to the superstructure deck in the form of a gantry crane over the stern could also be one way of handling anchors. Possibly some arrangement could also be made whereby the present freeboard deck would still be intact, but the superstructure deck could have a very large hatchway opening without closing appliances, i.e. it would form a bulwark, so that pipes stowed in such a space would then be less susceptible to filling with water because of their protected location.

In conclusion, it seems remarkable that in the now extensive experience of North Sea conditions very little serious attention seems to have been given to modifying existing designs to make them safer, apart from the fitting of longitudinal bulkheads.

AUTHOR'S REPLY

Initially, I wish to thank all those members who have contributed.

To Mr. J. McCallum

Additional trim aft induced whilst at sea at service speed would clearly increase the probability of shipping water over the stern. However, experimental evidence does not completely support the view that the cut away stern rapidly increases the tendency of the vessel to trim by the stern at service speed. Indeed, recent tank tests undertaken by the British Hovercraft Corporation on several supply vessel hull forms, indicate that the change of trim was very small and, surprisingly, generally occurred by the bow.

Clearly there is a need to obtain maximum buoyancy in the underwater portion of the aft body and this has now been recognized by both designers and operators. Loss of longitudinal and transverse stability is seen as only one part of the problem. In addition, there exists the problems of stern slamming, which is experienced when the vessel is stern first to the rig/platform, the sliding effect of the stern at large rudder angles, and lifting over the stern during anchor handling operations. In an attempt to reduce the effects of slamming there has been a trend in recent years towards 'V' shaped sections in the aft body and this represents the only major change in form which has occurred.

Without a radical change in concept and with the retention of ducts in the propulsion system the designer is, I believe, rather restricted regarding the question of additional buoyancy. The most he can do is to squeeze in the maximum amount of buoyancy by selecting the optimum construction technique. And in this respect it is thought the curved plate construction would be most suitable.

The desire for additional buoyancy has led to some interesting proposals. The fitting of a bulbous skeg is one such proposal. Another is the provision of raised side decks along the length of the cargo deck at its port and starboard extremities. Ewen & Kehela in their paper 'The Effect of Changes in Ship Dimensions on the Stability of Supply Vessels' showed that raised side decks can be an advantageous feature and suggested that this may be one way of improving the range of stability of existing vessels. This proposal seems to have met with some acceptance in the industry since a number of new supply vessels incorporate side tanks worked along the length of the cargo deck.

To Mr. L. Beckwith

The point you make, which as the paper states is specifically related to major foundering incidents, that the primary cause of casualty was in no case due to inadequate stability, is an extremely important one. This very fact coupled with the service experience to date would tend to contradict, superficially at least, the necessity for the formulation of intact stability standards specifically for supply vessels. Indeed, analysis of foundering casualties suggests that it is generally in the area of ship operational matters that standards should be improved.

Essential to the above conclusion is the meaning of the term 'inadequate stability' and is in itself a point which could be debated at some length. In the context of the paper it should be taken to mean failure to comply with IMCO Resolution A167 (ES.IV).

Whilst I agree that the IMCO stability criteria result from a pseudo-dynamic approach, I still believe my comment regarding environmental factors, viz. all hydrodynamic

forces induced by the sea, variation in buoyancy force, kinetic energy induced by the sea, energy dissipation etc., is still a valid statement. If you apply static criteria to ensure the safety of a vessel in what is a strictly dynamic environment then it follows the magnitude or degree of severity of those static criteria is such as to implicitly incorporate or encompass the effect of those environmental factors. It is analogous to the generally applied method of assessing ship structural strength. This is based, in most cases, on still water loads and occasionally quasi-static loads where adequate reserve strength for the additional dynamic loads is implicitly accounted for when fixing the stress levels.

It is worth noting whilst discussing criteria that if damage stability criteria are introduced the intact stability criteria will be of only secondary importance since, in general, damage standards will demand a higher degree of initial stability than required by intact standards.

The international conference at Strathclyde University to which you refer represents, to my mind, an important stage in the development of the concept and application of stability. It highlighted many basic questions which are essential to future progress. Perhaps most important of all, a need to precisely define both mathematically and non-mathematically what is meant by stability.

This may sound surprising and indeed quite straightforward after so many years of application to ships. It is, however, an extremely complex question, one which it may not be possible to answer until a more rationalized concept of stability has been formulated. It is clear from the proceedings of the conference that the new concepts will require a much deeper understanding of stability and perhaps a greater knowledge of mathematics than that required by the classical theory. This tendency raises the real danger that the gulf between the academic and the mariner will widen considerably. Like you, I hope the Society, by its membership of the IMCO working groups, will keep up to date with new developments but, in addition, I hope that the working groups prevent this gulf widening by bearing in mind the practical application of stability criteria, paying particular attention to the limitations of shipboard personnel.

The treatment of 'existing' ships subsequent to the introduction of international regulations always represents a difficult problem and will also prove so in the case of damage stability standards for supply vessels. In general, conventions tend only to deal with 'new' ships and national administrations may be left to assess this problem of existing ships on an individual basis. I do not believe there are many measures which could be undertaken without affecting adversely the operation of the vessel.

The provision of adequate hull fendering in way of those areas where damage could be critical to the survivability of the vessel is thought to be a reasonable 'stop gap' measure to recommend on existing ships. However, I believe such fendering should be permanently fixed to the hull and not of the portable inflatable type as you suggest. Fixed fendering can provide permanent protection above and below the W.L. whilst the inflatable type is limited to the water surface and its effectiveness is reliant on crew handling. An indirect way of tackling this problem is to ensure that all offshore installations served by supply vessels are provided with adequate leg fendering in way of mooring positions over the full range of possible operating draughts.

The danger of damage could also be reduced by the introduction of improved mooring systems. One system recently developed, which is an improvement over existing methods, includes both bow and stern mooring. The bow mooring consists of a spar buoy held under tension by a gravity base at the sea bed whilst stern mooring is provided by two lines led to the rig via 'anti-snatch' and 'weak link' devices, thus producing a three legged mooring system. Such a system, it is claimed by its designers, provides added safety against rig contact during cargo discharging operations and a more steady platform for cargo handling.

To Mr. M. B. Leese

There is little doubt that a supply vessel incorporating a 'boxed in' or fully enclosed cargo space as you suggest, when viewed from considerations of crew safety, stability and cargo protection, would be an attractive way of improving the apparent deficiences in existing designs. However, I do not believe that such an idea is likely to be adopted by the industry, principally because of the operational difficulties envisaged when handling cargo at sea.

The method of transferring cargo at the rig/platform is very crude and, in bad weather conditions, plagued with practical problems. Having personally witnessed such transfers in the North Sea during a force 7 (Beaufort scale) it became clear that the optimum design of cargo deck, operationally speaking, is one with the minimum amount of structural obstruction which, of course, is a feature of most existing designs. The three-dimensional movement of both the crane hook and the supply vessel can be large, thus the securing of items to be transferred to or from the hook is difficult and often dangerous. The lifting of cargo in way of an enclosed structure or through hatchways would, I believe, greatly increase this difficulty. Furthermore, the time spent discharging or loading at rigs/platforms could increase. This latter point is important since failure to operate successfully in the same sea and weather conditions as existing designs in what has become a very competitive industry could lead to doubts concerning the economic viability of such a 'boxed in' design.

It is quite likely that the length and hence size of supply vessels will increase in the future but not for reasons of safety. The trend towards exploration farther from shore has meant a corresponding increase in distances to supply, which may lead to a reduction in the number of round trips. For economic reasons, therefore, charterers may desire vessels of greater range and greater deadweight.

A number of new concepts have been proposed and for your information Fig. 1D indicates a design concept which was developed in Norway in 1974-5. As you can see, it is a substantial departure from existing designs. Features include a raised forecastle, a short towage deck aft, a covered cargo space with steel hatch covers folding aft, and Voith Schneider propulsion. This design incorporates some of the features you suggest but to my knowledge no supply vessels have been built accordingly.

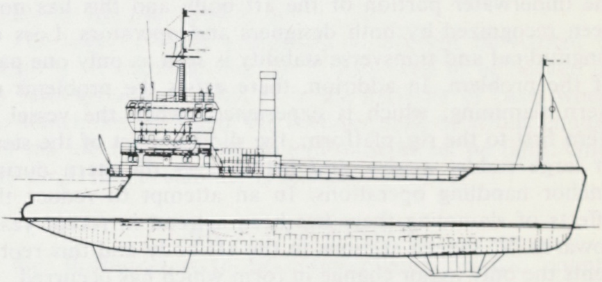


Fig. 1D

A new concept in supply vessel design.

To Mr. R. J. Bell

I note with interest your comments regarding pipe end protectors. I am familiar with the American Petroleum

Institute (API) specifications but these do not give any detailed design requirements for the configuration of end protectors and hence, in practice, their design varies widely. I would like to correct you on one point: API 5L requires a thread protector at only one end of the pipe and not at both ends as you suggest, a coupling being required at the other end. Another point is that not all pipes are threaded at the ends—line pipe for instance may be bevel-ended in preparation for welded joints. For your information the following rough sketches (Figs. 2D, 3D, 4D, 5D, 6D, & 7D) indicate typical designs of end protectors which are presently used in the pipe manufacturing industry when working to API specifications 5A & 5L. It will be noted from Figs. 4D & 5D that the protectors, in the case of line pipe for use in Arctic areas, are totally enclosed at the ends of the pipe. This is done to keep the bore of the pipe clear of packed snow and ice which would otherwise interfere with running operations when the pipe is put into use. To my knowledge this represents the only real attempt to secure watertightness whilst all other types (and these will be used in the majority of operational areas) are concerned only with thread or bevel end protection.

I would agree that it may be a simple task to physically modify existing end protectors to also act as watertight blanks or caps, but I do not believe it would be a simple task to achieve this in the organizational sense, because of the variation of practice which exists in the industry and the increased costs which will inevitably be involved.

Of course, even if consistency of practice could be achieved, the blanking of pipe cargo does not solve completely the problem of water entrapment. This measure will prevent water entry into pipes but not between pipes, not between pipes and bulwarks, and not between pipes and other types of cargo.

With regard to your second point there is little doubt that Rahola's contribution in 1939 in the field of ship stability was a significant one and of great value in the formulation of workable stability criteria. As I have mentioned in the paper, the results of his investigations form the basis of IMCO Resolution A167 (ES.IV) which was published in 1968 and also the basis of several national criteria.

Clearly these criteria provide a simple yardstick for assessing and comparing ship stability but are based upon a purely 'statical' approach which includes many obvious assumptions. In recent years there has been a desire in research circles to place the science of ship stability on a more rational basis which would reflect the ship's motion characteristics and the environmental conditions in which a ship may operate during its service life. This is not merely a revision of Rahola's work as you intimate, but a radical change in philosophy or, if you wish, a change in direction. It is an exciting development, one which I personally welcome.

Clearly Surveyors cannot play an active role in this research and development since it is not the Society's function to do so, although you may think we have the technical capability. The Society's function in the field of stability is to apply the requirements of those governments or national authorities on whose behalf the Society may be acting. Certainly it is an advantage for the Surveyor to be aware of scientific and technological advances in ship stability but, as I have implied, he must be predominantly concerned with application. This is no mean task, indeed it is an important function. Approval of stability information represents the penultimate link in the chain before reaching the onboard user and it is at this stage that the Surveyor has much to contribute.

It is more than probable, as technology advances in the new direction, that stability information will become significantly more complex, requiring a far deeper understanding of the subject. As I suggested to Mr. Beckwith this may create an even greater gulf between the scientist and the mariner. Clearly then, in such a situation, for Surveyors to ensure that stability information placed aboard the ship is simple to use and understand, will be achieving a great deal.

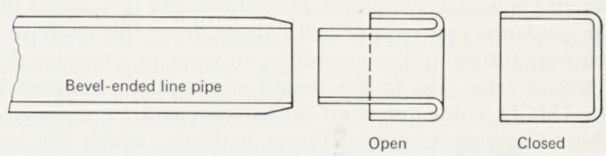


FIG. 2D

Plastic end cap (held on by dimensional tolerance in design).

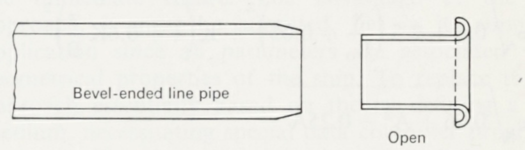


FIG. 3D

Steel bevel ring (held on by eye and peg arrangement).

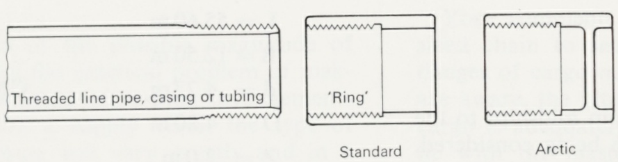


FIG. 4D

Steel protector for pin (uncoupled or male) end.

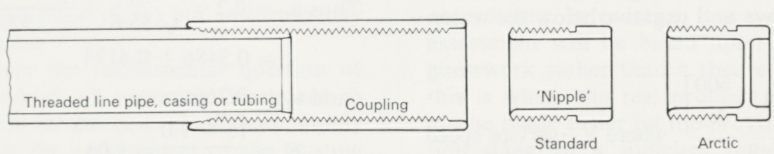


FIG. 5D

Steel protector for box (coupled or female) end.

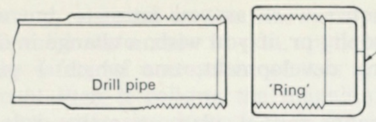


FIG. 6D

Steel protector for pin (male) end.

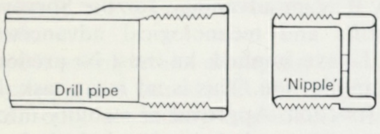


FIG. 7D

Steel protector for box (female) end.

TO MR. G. A. SMITH

With regard to your first point, guidance as to the magnitude of accelerations likely to be experienced by supply vessels or, indeed, small ships in general is not readily extractable from the Society's Rules. This is often 'hidden' in a relationship between design loads, permissible stress and the structural model used. Recently, however, there has been a move by the Society to standardize ship structure design loads and calculation procedures and it is hoped that the guidance you require will emerge from this exercise in the near future. In the meantime, to serve as a rough guide, I would refer you to the design accelerations adopted by the IMCO 'Code for the Construction and Equipment of Ships Carrying Liquefied Gases in Bulk', which for convenience are repeated here:—

$$a_z = \pm a_o \sqrt{1 + \left(5.3 - \frac{45}{L_o}\right)^2 \left(\frac{x}{L_o} + 0.05\right)^2 \left(\frac{0.6}{C_b}\right)^{1\cdot5}}$$

$$a_y = \pm a_o \sqrt{0.6 + 2.5\left(\frac{x}{L_o} + 0.05\right)^2 + K\left(1 + 0.6K\frac{z}{B}\right)^2}$$

$$a_x = \pm a_o \sqrt{0.06 + A^2 - 0.25A}$$

$$\text{where } A = \left(0.7 - \frac{L_o}{1200} + 5\frac{z}{L}\right)\left(\frac{0.6}{C_b}\right)$$

L_o = Rule length of ship.

C_b = Block coefficient.

B = Moulded breadth.

x = Longitudinal distance (m) from midships to the centre of gravity of the item being considered. x is positive forward of amidships, negative aft of amidships.

z = Vertical distance (m) from the ship's actual waterline to the centre of gravity of the item. z is positive above and negative below the waterline.

$$a_o = 0.2\frac{V}{\sqrt{L_o}} + \frac{\left[34 - \frac{600}{L_o}\right]}{L_o}, \text{ where V = service speed in knots}$$

$$K = \frac{13GM}{B}, \text{ where } K \geqslant 1.0 \text{ and GM = metacentric height}$$

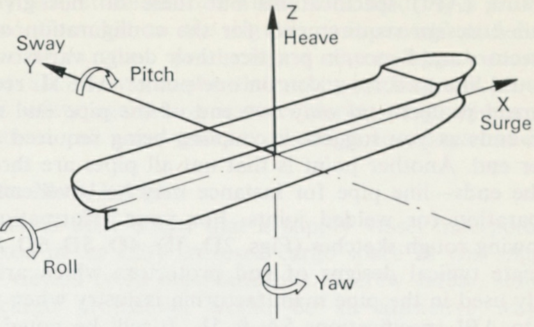
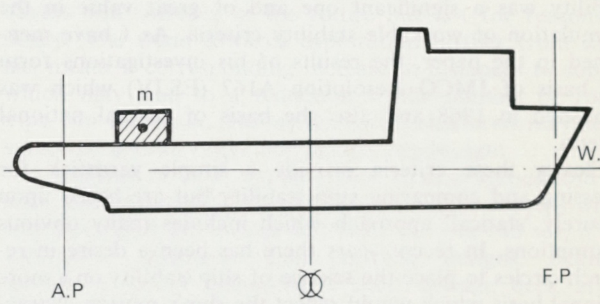


FIG. 8D

Reference system.

These expressions, which are based upon a probability level of 10^{-8} (once in 20 years) in the North Atlantic, represent the maximum dimensionless accelerations (i.e. relative to the acceleration due to gravity 'g') in the three primary directions (*see* Fig. 8D) and incorporate the effects of surge, sway, heave, roll, pitch and yaw. The expression for a_y includes the component of static weight in the transverse direction due to rolling and a_x includes the component of the static weight in the longitudinal direction due to pitching. It should be appreciated that these expressions are specifically relevant to vessels greater than 50 metres in length and of a different hull form, thus they must be applied with some discretion.

Clearly the above expressions are dependent upon a number of variables which will be different for each case, however, for guidance in obtaining an impression of the magnitude of these accelerations the following example is considered. A typical supply vessel with 10 tonnes of cargo placed aftermost and extreme port side.



Example loading.

L = 55.60 m	Δ = 1838 T
B = 12.50 m	V = 13.0 knots
d = 4.27 m	GMT = 1.0 m
D = 4.80 m	C_b = 0.60
Z = 2.0 m	m = 10.0 tonnes
X = −22.0 m	

$$\text{Thus } a_o = 0.2 \times \frac{13.0}{\sqrt{55.6}} + \frac{\left[34 - \frac{600}{55.6}\right]}{55.6}$$

$$= 0.3486 + 0.4174$$

$$\text{and } a_o = 0.766$$

$$K = \frac{13 \times 1.0}{12.50} = 1.04$$

$$A = \left(0.7 - \frac{55.60}{1200} + \frac{5 \times 2.0}{1200}\right)\left(\frac{0.6}{0.6}\right)$$

Thus A = 0.8336

Combined accelerations: —

$$a_z = 0.766\sqrt{1 + \left(5.3 - \frac{45}{55.6}\right)^2\left(-\frac{22.0}{55.6} + 0.05\right)^2\left(\frac{0.6}{0.6}\right)^{1.5} + \left(0.6 \times 1.04^{1.5} \times \frac{5.0}{12.5}\right)^2}$$

thus $a_z = 1.414\text{g m/s}^2$

$$a_y = 0.766\sqrt{0.6 + 2.5\left(-\frac{22.0}{55.6} + 0.05\right)^2 + 1.04\left(1 + 0.6 \times 1.04 \times \frac{2.0}{12.5}\right)^2}$$

thus $a_y = 1.125\text{g m/s}^2$

$$a_x = 0.766\sqrt{0.06 + 0.8336^2 - 0.25 \times 0.8336}$$

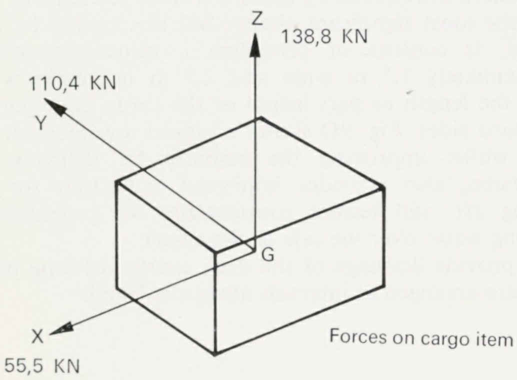
thus $a_x = 0.566\text{g m/s}^2$

Combined static and dynamic forces: —

Normal to deck (Z) $= m \times a_z \times g$
$= 10.0 \times 1.414 \times 9.81$
$= 138.8$ KN

Tangential to deck in transverse direction (Y) $= m \times a_y \times g$
$= 10.0 \times 1.125 \times 9.81$
$= 110.36$ KN

Tangential to deck in longitudinal direction (X) $= m \times a_x \times g$
$= 10.0 \times 0.566 \times 9.81$
$= 55.5$ KN



Forces on cargo item

Of course, a knowledge of the possible magnitude of accelerations does not make the practical problem of making an 'on the spot' assessment of lashing arrangements entirely straightforward. On a supply vessel the type of cargo and method of lashing will vary greatly and in a random way, thus making it difficult to accurately distribute theoretically calculated loads in the wires or chains. If all deck cargo was in containerized form then a standard method of securing deck cargo could be devised, as in the case of deck loads on container ships, but unfortunately this is not the case in practice.

Your second point raises the fundamental question of whether the existing method of assessing freeboard is applicable or indeed logical in the case of offshore supply vessels. Clearly you favour the replacement of the existing method by one relating freeboard to deck wetness, a view held by a large number of people for application to ships in general. At the time of the 1966 International Conference on Load Lines many believed that wetness was an important factor and should have received more consideration than it evidently did receive. The results of much research work were available at the time, in particular that carried out by Newton and Goodrich.

In principle, I would agree with you that the idea of relating freeboard to the probability of wetness is a more logical approach to assessing freeboard than that in current use. However, whilst possessing obvious merits, I do not see this approach being introduced until many of the assumptions adopted in the theoretical basis have been confirmed by service experience and all the variables have been clearly defined and accounted for.

Briefly, this approach involves the determination of the vessel's relative vertical motion with respect to the water surface by computing the vectorial sum of the displacements due to heave and pitch and the wave displacement in some standardized sea wave spectrum. When the relative displacement exceeds the local freeboard at some point along the vessel's length then the shipping of water or some degree of wetness occurs. The probability that the relative displacement exceeds a certain value may then be determined by using some recognized distribution, e.g. a Rayleigh distribution. The 'strip' theory, which is currently used to compute a vessel's response in regular waves, embodies a number of assumptions which make the applicability to supply vessels doubtful. For example, the theory assumes linear response, a wall sided vessel and relatively small motions. In supply vessels the pitch motion may be large, thus affecting the wall-sided assumption and in turn causing non-linearity in response. This raises the question of whether the theoretical solution should be correlated with full scale tests before application. Many other questions would have to be decided upon, including the relevant sea states, speed, headings, an acceptable level of probability of wetness and, not the least, a comprehensive definition of wetness.

Obviously this approach is substantially more complex than the existing method and this fact, coupled with the outstanding questions on wetness, is going to make, I believe, any international agreement difficult to achieve in the immediate future. The advantage of the existing approach, it must be admitted, lies in its simplicity of application since all parameters are associated with the geometrical properties of the ship. To replace this by an approach essentially based on theory defining a random medium, necessitating special data computer programs and analysis will not endear authorities to make proposals for rapid change in the load line rules.

To Capt. V. Anderson

Your comments regarding the use in practice of undersized chain to secure deck cargo highlight the potential danger of cargo movement whilst at sea. As no doubt you are aware, the ultimate responsibility for ensuring that the cargo is adequately secured for the intended voyage must lie with the master of the vessel. His assessment must, therefore, take account of all the environmental forces which are likely to occur, i.e. dynamic forces due to the motion of the vessel, gravitational forces, sea impact forces, wind pressure etc., and must select the lashing arrangement accordingly. In practice, it is more than probable that this assessment will be based upon experience or a degree of guesswork rather than a theoretical approach and perhaps this is where the real problem originates. Clearly his task can be made easier by the provision, in the vessel's construction stage, of a sufficient number of deck and bulwark anchoring points Furthermore, the provision of wires or chains with an adequate factor of safety against failure loads.

In general, I would agree with you that stanchions should be supplemented by chains or wires as a means of preventing the movement of cargo on deck. Items of cargo, particularly those stowed near the aft end and outboard, will be subjected to high vertical accelerations in addition to horizontal accelerations and, therefore, both means will be needed to prevent movement.

You are quite right to say that stability can be seriously affected by cargo movement. The latest major casualty (not included in the paper) illustrates this point. It concerns a supply vessel of typical proportions which was taking on pipe from a drilling rig in the North Sea whilst a 4 metre swell and 6 knot wind persisted from an abeam direction (port). The vessel was moored stern to rig, one line over the stern to the rig leg and one line over the bow to an anchor. The cargo deck was virtually full of pipes up to the top of the cargo rails, some 200 pipes amounting to 160–180 tons. More pipe was about to be loaded when the swell induced a rolling motion to starboard which caused about half the cargo to roll from the port side over the starboard pipe cargo and into the sea. A large angle of heel to starboard resulted and efforts were made to reduce this by pumping dry the flume stabilization tank. However, such action was not effective and it was decided to cut the vessel loose and to allow it to drift. Further swell action induced a greater angle of heel, shipping of water took place and the vessel eventually capsized and sunk, bow up in 485 ft of water. Casualties of this nature remind us of the importance of adequate securing arrangements both during and after loading cargo.

To Mr. J. Crawford

You refer to my experience in the North Sea and I can confirm that it has proved of immense value in gaining an understanding of some of the operational problems connected with the assessment of stability on these vessels. Clearly such problems cannot be readily or fully comprehended at an office desk remote from the subject.

In general, action taken by classification societies up to the present time has been limited to the introduction of structural rules, although in the case of Det norske Veritas their rules also include standards of intact and damage stability which must be complied with. However, through membership of the International Association of Classification Societies (IACS), which plays an observer's role on the IMCO sub-committee on subdivision, stability and load-lines, it has been possible for classification societies to assist and contribute on all matters affecting the seaworthiness of supply vessels.

I note you raise the age old argument regarding the use of the words vessel or ship, and clearly you prefer the designation supply ship to the generally used supply vessel. If you study all the available literature on the subject you will find each designation being used as often as the other. To clarify matters, I would draw your attention to the definition adopted by the International Regulations for Preventing Collisions at Sea, 1972 viz., 'the word "vessel" includes every description of water craft, including non-displacement craft and sea planes, used or capable of being used as a means of transportation on water'. If you accept this definition then the 'ship' is a particular type of 'vessel' and it follows that a supply ship is a vessel and, therefore, your preference, I would agree, appears to be technically correct. However, you must not be too harsh on those of us who cannot see the difference and carry on using a hybrid designation, such as supply vessel.

To Mr. O. M. Clemmetson

I am familiar with the article to which you refer. It appeared in the Lloyd's List on 16th May 1973, the article being headed 'Marine Support Operations to the Offshore Oil Industry' and written by Captain P. King. For reasons given to Mr. Leese, it is unlikely that the design of supply vessels will change radically in the near future. In my opinion the shelter deck supply vessel could only operate successfully in good environmental conditions and with an effective mooring system. These two factors would be particularly difficult to consistently achieve in the North Sea and this probably reflects the reluctance of European designers to introduce radical changes in design.

The side tank concept mentioned in my reply to Mr. McCallum, introduced by Messrs. Talbot & Jackson, represents the most significant change but this cannot be called radical. It consists of providing a watertight structure approximately 1.5 m wide and 2.5 m in height worked along the length or part length of the cargo deck port and starboard sides. Fig. 9D shows a typical arrangement. This idea, whilst improving the stability by delaying edge immersion, also provides improved protection for crew moving aft, and lessens considerably the probability of shipping water over the side of the vessel.

To provide drainage of the deck overboard long narrow ports are arranged at intervals along the length.

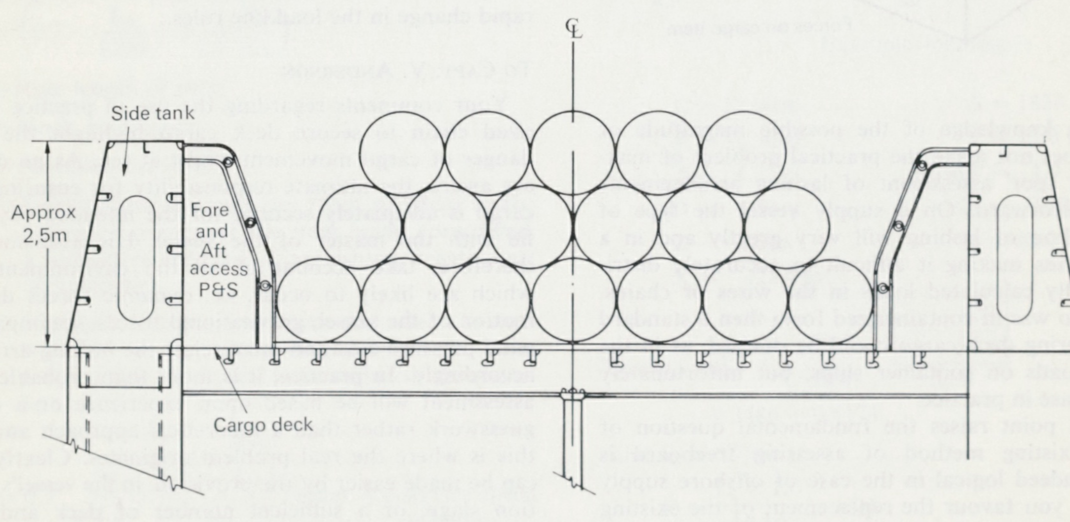


Fig. 9D

Typical arrangement of cargo deck with side tanks.

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